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Sensitivity Analysis of Vehicle Tripped Rollover Model

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16. Abstract

The time domain sensitivity analysis of System Technology Incorporated's (STI) Tripped Rollover Vehicle Model has been conducted utilizing sensitivity programs developed by the University of Missouri-Columbia. The eight degree of freedom STI tripped rollover model simulates the motion of a vehicle which skids laterally on pavement and then impacts with a curb.

The sensitivity of vehicle system parameters and relevant system response variables have been investigated and ranked in order of importance in tripped rollover situations using percentage sensitivity functions and various sets of initial conditions. Based on energy considerations, vehicle rollover prevention energy reserve (RPER) has been derived and used as an indication of vehicle rollover stability. The results reveal that the minimum value of RPER approaches zero at the threshold of rollover and if this minimum falls below zero then vehicle rollover will occur. The sensitivity results show that vehicle geometric and wheel-suspension deformation parameters are by far the most influential characteristics in a tripped rollover situation, followed by vehicle's mass properties. The influence of vehicle stiffness and damping parameters on vehicle RPER was found to be quite small in tripped rollover maneuvers.

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1.0 INTRODUCTION

In the past few years, the National Highway Traffic Safety Administration (NHTSA) has displayed a great deal of interest in determining the underlying causes of vehicle rollover accidents and developing vehicle and roadway design criteria which will reduce both the number and severity of rollover accidents. Vehicle rollover is expected to be a growing problem as NHTSA 1986 accident data files show that utility vehicles and small pickup trucks have the greatest number of rollover deaths per million vehicles (155.7 and 127.7 respectively), while also having the highest percentage of fatal accidents involving rollovers (66% for utility vehicles and 45% for small pickups). At the same time, sales of these types of vehicles have been growing dramatically with the market share of utility vehicles increasing from 2% to 6% in the past decade.

There are two types of vehicle rollovers which occur frequently in single vehicle accidents. A tripped vehicle rollover can result if the vehicle strikes a curb or other obstacle. An untripped vehicle rollover occurs during an abrupt vehicle maneuver without the help of a tripping mechanism. Both tripped and untripped vehicle rollovers are equally dangerous and detailed investigations of their behavior are needed in order to determine their causes and thereby reduce vehicle rollover propensity.

Several projects have been initiated to study the factors involved in rollover accidents. Calspan Corporation studied the rollover potential of vehicles on embankments, sideslopes, and other roadside features on behalf of the Federal Highway Administration [1]. The Tripped Rollover Vehicle Simulation [2] developed by Systems Technology, Inc. (STI) is being used by NHTSA to predict vehicle rollover response during maneuvers involving lateral impact with a rigid curb. The state of Maryland is currently collecting accident data which will be used to determine the statistical correlation of factors associated with vehicle rollover, and the University of Missouri-Columbia is currently developing an Advanced Vehicle Rollover Model to study the dynamic behavior of vehicles in both tripped and untripped vehicle rollovers.

The STI Tripped Rollover Vehicle Simulation can be used to predict a vehicle's response during a maneuver in which the vehicle slides laterally into a rigid curb with some small heading angle. The STI simulation allows the user to select different sets of initial conditions such as lateral vehicle speed, angle of impact with the curb, initial distance from the curb, ect. The model has eight degrees of freedom which include vertical and horizontal translations, roll, pitch, and yaw. While this model generates vehicle response in tripped rollover situations, the influence of various parameters on rollover propensity is not immediately evident.

Knowledge of how the different characteristics describing a vehicle's dynamic systems and sub-systems affect its behavior in accident situations is an essential aspect of understanding the rollover problem. However, based on simulation runs, it is not always clear which characteristics have the greatest influence on the behavior of a complex mechanical system such as a vehicle. Also, attributes which may be important in one situation may be unimportant under different conditions, making the entire analysis even more complex. It has been shown [3-8] that sensitivity methods are very useful in determining the influence of various design, operational, and environmental parameters on the response of complex mechanical systems. As a result, it was proposed that the sensitivity algorithms developed by the University of Missouri-Columbia be applied to STI's Tripped Rollover Vehicle Simulation to gain insight into the influence of certain parameters on tripped vehicle rollover behavior.

2.0 PROJECT OBJECTIVES

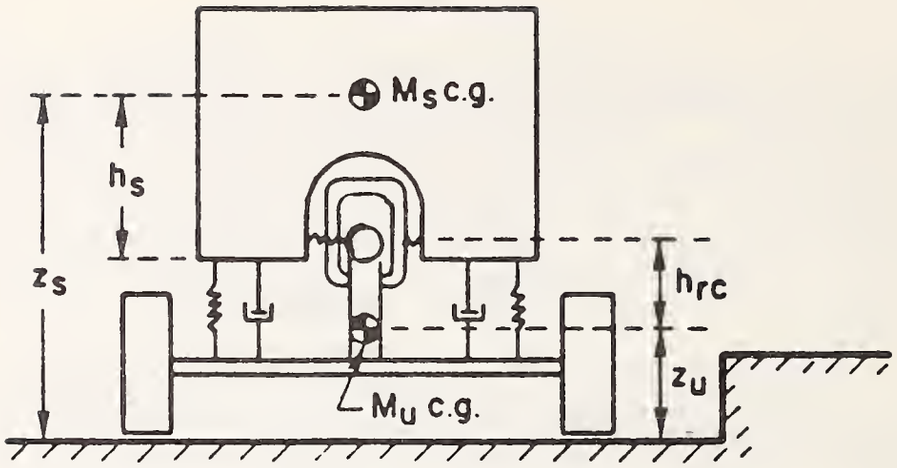
The primary objective of this project was to conduct a sensitivity analysis of STI's Tripped Rollover Simulation. The sensitivity of vehicle system parameters were investigated and ranked in order of importance in tripped rollover situations.

Relevant system response variables such as sprung and unsprung mass roll angles, roll velocities, lateral velocities, ect. were analyzed using percentage sensitivity functions. The percentage sensitivity function was chosen to provide the clearest physical interpretation of results. In addition to system response variables, an additional function, vehicle rollover prevention energy reserve (RPER), was derived based on energy considerations and was used as an indication of vehicle rollover stability. The sensitivity of this function to system parameter changes was then conducted using various sets of initial conditions.

3.0 STI TRIPPED ROLLOVER MODEL

The STI tripped rollover model [2] simulates the motion of a vehicle which skids laterally on pavement and comes into contact with a roadside curb. The model simulates the vehicle's skidding motion prior to curb impact, as well as vehicle motion after impact. Vehicle rollover is assumed to take place when the roll angle of the unsprung mass exceeds approximately 85 degrees.

The simulation uses a non-linear eight degree of freedom model which utilizes simple linear subsystems to model tires, suspension and impact force. The vehicle model (Fig. 1, 3 and 4) consists of two masses, a sprung mass representing the vehicle chassis and a single unsprung mass which represents the combined front and rear suspension systems. The sprung and unsprung masses have separate degrees of freedom for roll, lateral and vertical translation, while vehicle yaw and pitch are analyzed using a single lumped mass. The impact force for lateral wheel/curb impact is modeled using both plastic and elastic deformations. Damping effects are included through energy dissipative forces in the vehicle's tires, the lateral bushing between the sprung and unsprung masses, the shock absorbers in the suspension system, and the wheel/curb impact force.



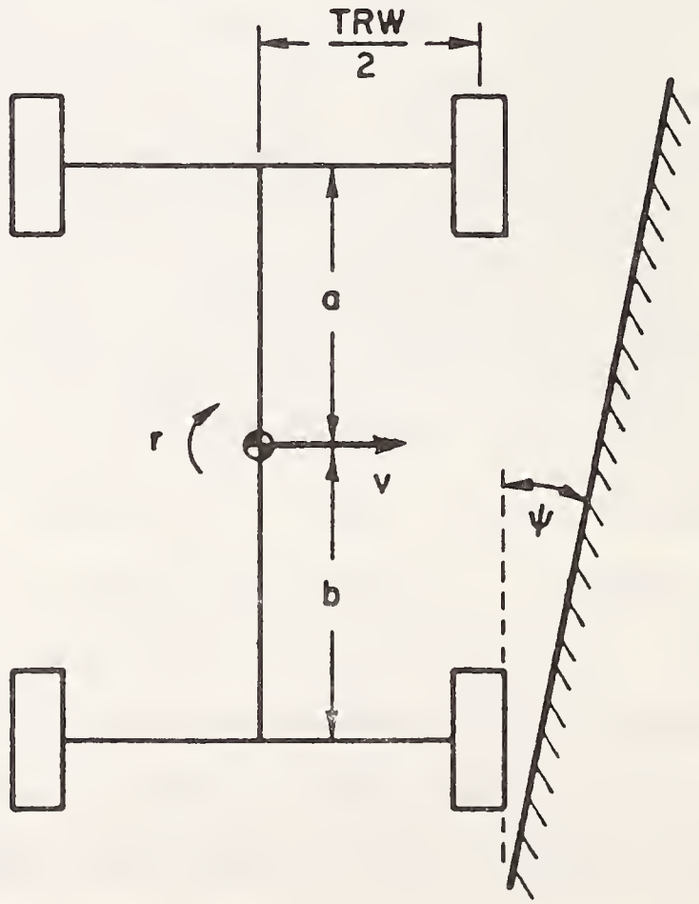


Fig. 1

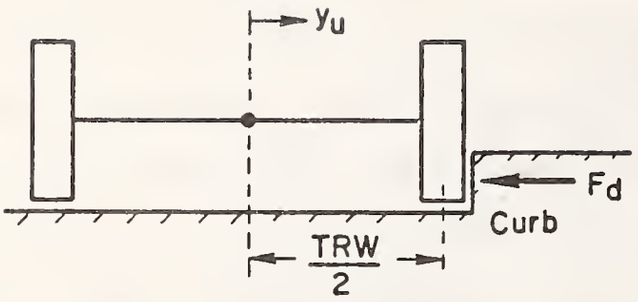
3.1 Tire Model

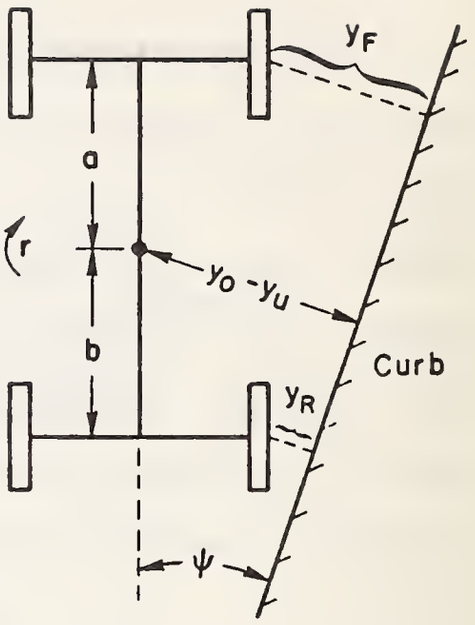
The tire normal reactions (F_{ZLF}, F_{ZRF}, F_{ZLR}, F_{ZRR}) are based on a linear tire deformation model. The tire normal forces are proportional to tire deflections caused by vehicle heave, pitch and roll. The tire deflections are determined by the finding the difference between the undeflected tire radius and the distance between the axle and the road surface under each wheel. Tire damping forces are also included and are proportional to the tire rates of deformation.

Tire lateral sliding forces (F_{YLF}, F_{YRF}, F_{YLR}, F_{YRR}) are functions of tire normal forces, tire/surface frictional properties, and sliding velocity.

3.2 Suspension Model

The vehicle suspension model consists of two springs which have upper and lower bump stops. The springs deflect as the roll angles of the sprung and unsprung masses change, and as the two masses heave in relation to each other. The bump stops prevent excessive roll angle and heave differences between the two masses. Damping in the suspension is included to represent both shock absorbers and bump stop energy dissipation.





y_o = initial conditions for starting distance of m_u c.g. from curb

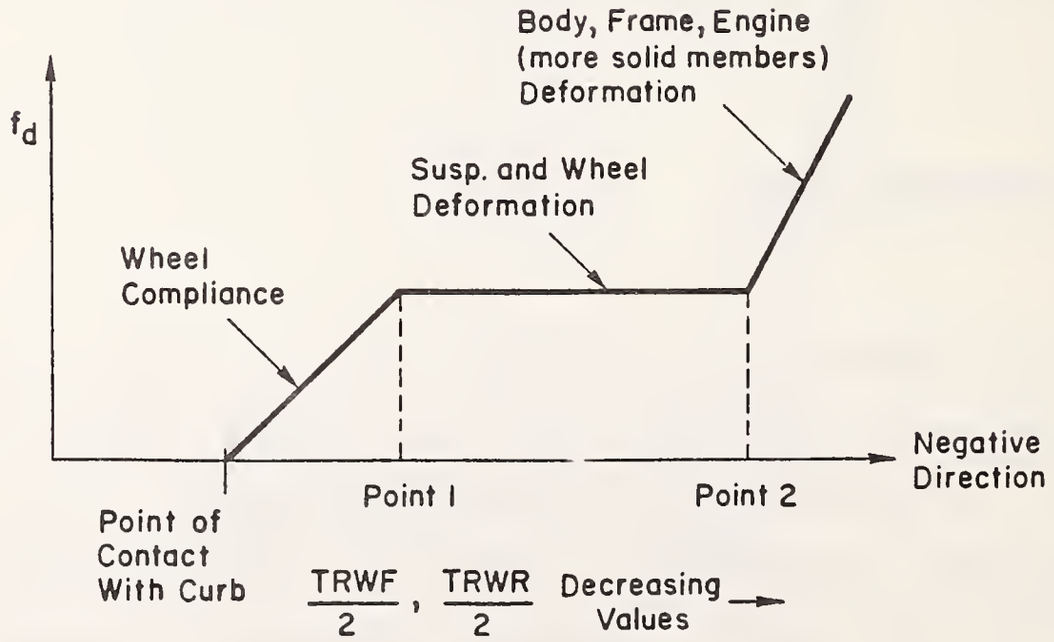


Fig. 2

3.3 Impact Force

The wheel/curb impact forces (F_{dR}, F_{dL}) are dependent on the amount and rate of deformation caused by curb impact. The impact force is modeled using two linear elastic regions and one region of plastic deformation (Fig. 2). If the vehicle rebounds, the work done by the deformation force across the first and third regions is recoverable, but the work done across the second region is lost. The impact forces are calculated separately for the front and rear wheel/suspension systems.

3.4 Equations of Motion

The sprung and unsprung masses are analyzed separately for six of the model's eight degrees of freedom. The equations of motion for each masses' roll angle (ϕ_s, ϕ_u), lateral position (y_s, y_u) and vertical position (Z_s, Z_u) were obtained by applying Newton's second law of motion to the sprung mass (Fig. 3) and unsprung mass (Fig. 4).

The forces which act on the sprung mass include gravity, the two forces of the suspension springs and dampers (F_{SL} and F_{SR}), and a pin force at the bushing which connects the sprung and unsprung masses (point F_1 in Fig. 3 and 4). The sprung mass roll moment equation follows:

$$I_s \ddot{\phi}_s = (F_{SL} - F_{SR}) \cdot s + F_1 \cdot h_{F_1}$$

The equations of sprung mass lateral and vertical motions are written:

$$M_s \ddot{Y}_s = (F_{SL} + F_{SR}) \cdot \sin(\phi_s) - F_1 \cdot \cos(\phi_s)$$

$$M_s \ddot{Z}_s - M_s g = -(F_{SL} + F_{SR}) \cdot \cos(\phi_s) - F_1 \cdot \sin(\phi_s)$$

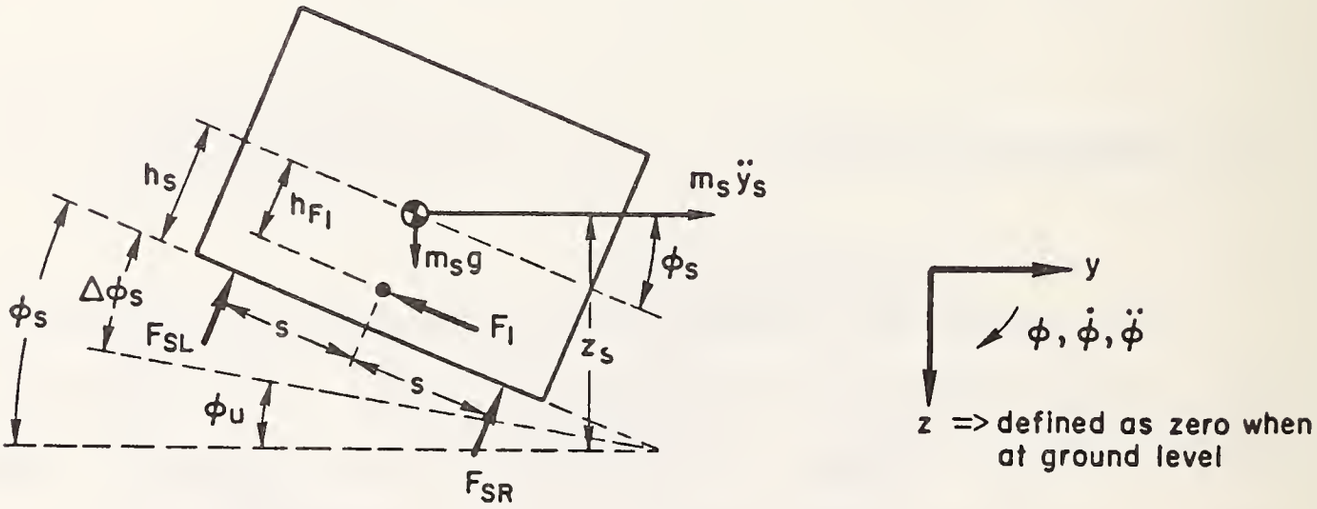


Fig. 3

The forces which act on the unsprung mass include gravity, the suspension spring and damping forces, the bushing pin force, as well as tire lateral friction forces (F_{YL}, F_{YR}), tire normal reactions (F_{ZL}, F_{ZR}), and the curb impact force (F_d) . The equations of motion for unsprung mass roll, lateral and vertical motions are obtained respectively:

$$I_u \ddot{\phi}_u = -F_{SL} \cdot s \cdot \cos(\Delta\phi_s) + F_{SR} \cdot s \cdot \cos(\Delta\phi_s) + F_{ZL} \cdot y - F_{ZR} \cdot x$$
$$- F_{YL} \cdot z_u - F_{YR} \cdot z_u + F_1 \cdot h_{rc} \cdot \cos(\Delta\phi_s) - F_d \cdot (Z_u + H_{curb})$$

$$M_u \ddot{Y}_u = -(F_{SL} + F_{SR}) \cdot \sin(\phi_s) + F_1 \cdot \cos(\phi_s) - F_{YR} - F_{YL} - F_d$$

$$M_u \ddot{Z}_u - M_u g = (F_{SL} + F_{SR}) \cdot \cos(\phi_s) + F_1 \cdot \sin(\phi_s) - F_{ZR} - F_{ZL}$$

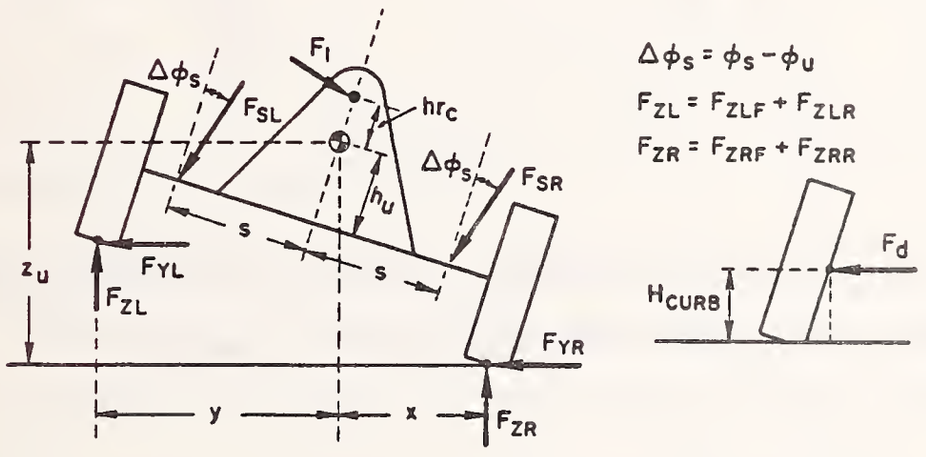


Fig. 4

The yaw equation of motion of the vehicle includes the individual tire lateral forces (F_{YLF}, F_{YLR}, F_{YRF}, F_{YRR}) and curb impact forces separately for each axle (F_{dF}, F_{dR}), (Fig. 5). The yaw equation of motion includes a dynamic coupling between sprung mass roll, vehicle pitch, and vehicle yaw, and follows:

$$I_z \dot{r} = -(F_{YLF} + F_{YRF} + F_{dF}) \cdot a + (F_{YLR} + F_{YRR} + F_{dR}) \cdot b + \dot{\phi}_s \dot{\theta} (I_x - I_y)$$

Vehicle pitch is treated as a single motion for the entire vehicle (Fig. 6). The only forces affecting vehicle pitch are the individual normal tire reactions (F_{ZLF}, F_{ZRF}, F_{ZLR}, F_{ZRR}).

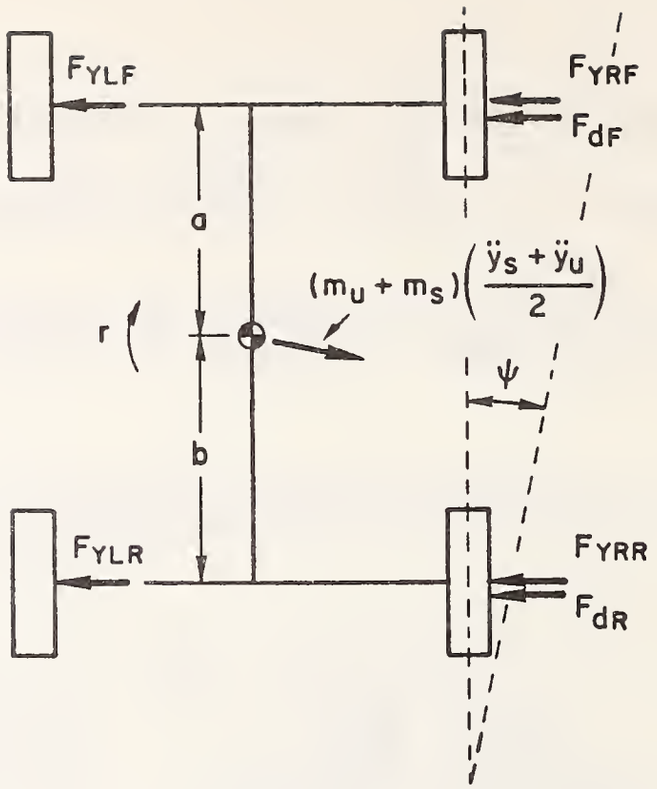


Fig. 5

The pitch equation of motion includes a dynamic coupling between vehicle yaw, pitch, and sprung mass roll angle:

$$I_y \ddot{\theta} = (F_{ZLF} + F_{ZRF}) \cdot a - (F_{ZLR} + F_{ZRR}) \cdot b + \dot{\phi}_s \, r \, (I_z - I_x)$$

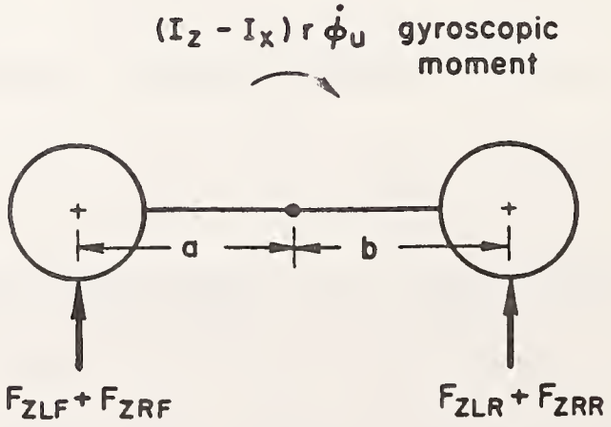


Fig. 6

3.5 Modifications to STI's Tripped Rollover Model

An inaccuracy was discovered in an assumption made by STI in their tripped rollover model. Several small angle assumptions were made affecting the sliding distance of the vehicle prior to curb impact thereby affecting the curb impact velocity. For example, the distance that the vehicle skids before impact should be equal to the distance between the vehicle and the curb measured in the direction of vehicle motion. Under STI's reference system, the distance the vehicle skids before impact is always measured perpendicular to the curb, which is a shorter distance in cases with a yaw angle. While STI's model was reasonably accurate at small heading angles, it was deemed necessary to add an non-inertial reference system (NIRS) to the model (Fig. 7). The addition of the NIRS ensures that the distance the vehicle skids before impact is perpendicular to the vehicle (i.e. in the direction of lateral velocity).

For the purposes of sensitivity analysis the STI simulation was rearranged so the absolute position of the vehicle relative to the curb was measured from the front right tire. This was done to ensure that the time of impact would be independent of vehicle track width. If the absolute position of the vehicle was measured from the vehicle center of gravity position (as it was in the original simulation) then the distance traveled prior to impact by a vehicle with a narrow track width would be longer than that traveled by a vehicle with a wide track width. Conse-

quently the two vehicles would hit the curb at slightly different times, skewing the sensitivity results. Measuring the vehicles absolute position from the front right tire guarantees that the distance traveled prior to impact is independent of the vehicle track width, which in turn guarantees identical impact times.

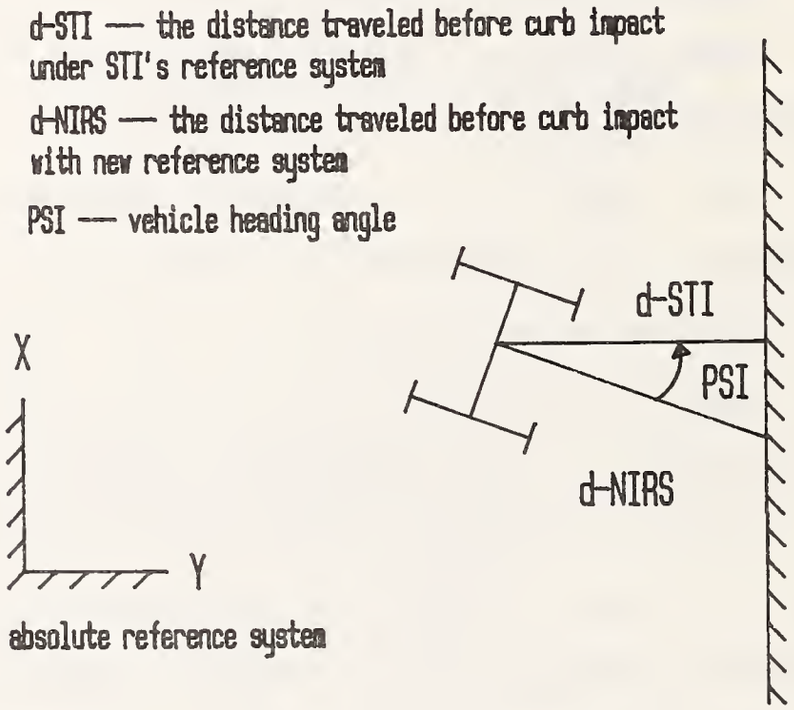

d-STI — the distance traveled before curb impact under STI's reference system
d-NIRS — the distance traveled before curb impact with new reference system
PSI — vehicle heading angle
X
Y
absolute reference system
d-STI
PSI
d-NIRS

Fig. 7

3.6 Validation of Fortran Subroutine

In order to perform a sensitivity analysis on STI's tripped rollover model it was necessary to translate STI's Basic program into an equivalent Fortran subroutine. The Fortran version of STI's model was verified by comparing vehicle response and energy using several sets of initial conditions. The vehicle data used

throughout this project represents a 1975 Chevy Nova and is listed in Appendix A. The system response obtained from the original STI simulation and the Fortran subroutine were nearly identical over the relevant time period (Fig. 8 - 9). Small differences appeared towards the end of the rollover simulation when vehicle rollover was nearly complete. The energy analysis also had comparable results (Fig. 10 - 11). We attribute any discrepancies between the results obtained using the STI's Basic simulation and UMC's Fortran version to differences in the methods used to integrate the differential equations of motion. The model was validated using STI's reference system which includes the inaccuracies mentioned in section 3.5, however, the change in the reference system should not affect the system response when the heading angle was zero. Graphs showing system response obtained using the Fortran model and 1975 Chevy Nova data are shown in Appendix C. The initial conditions (listed in Appendix B) represent a vehicle which is sliding sideways toward a curb at a speed of 23.075 Ft/S. The front wheel is initially located 7.5 Ft. away from the curb.

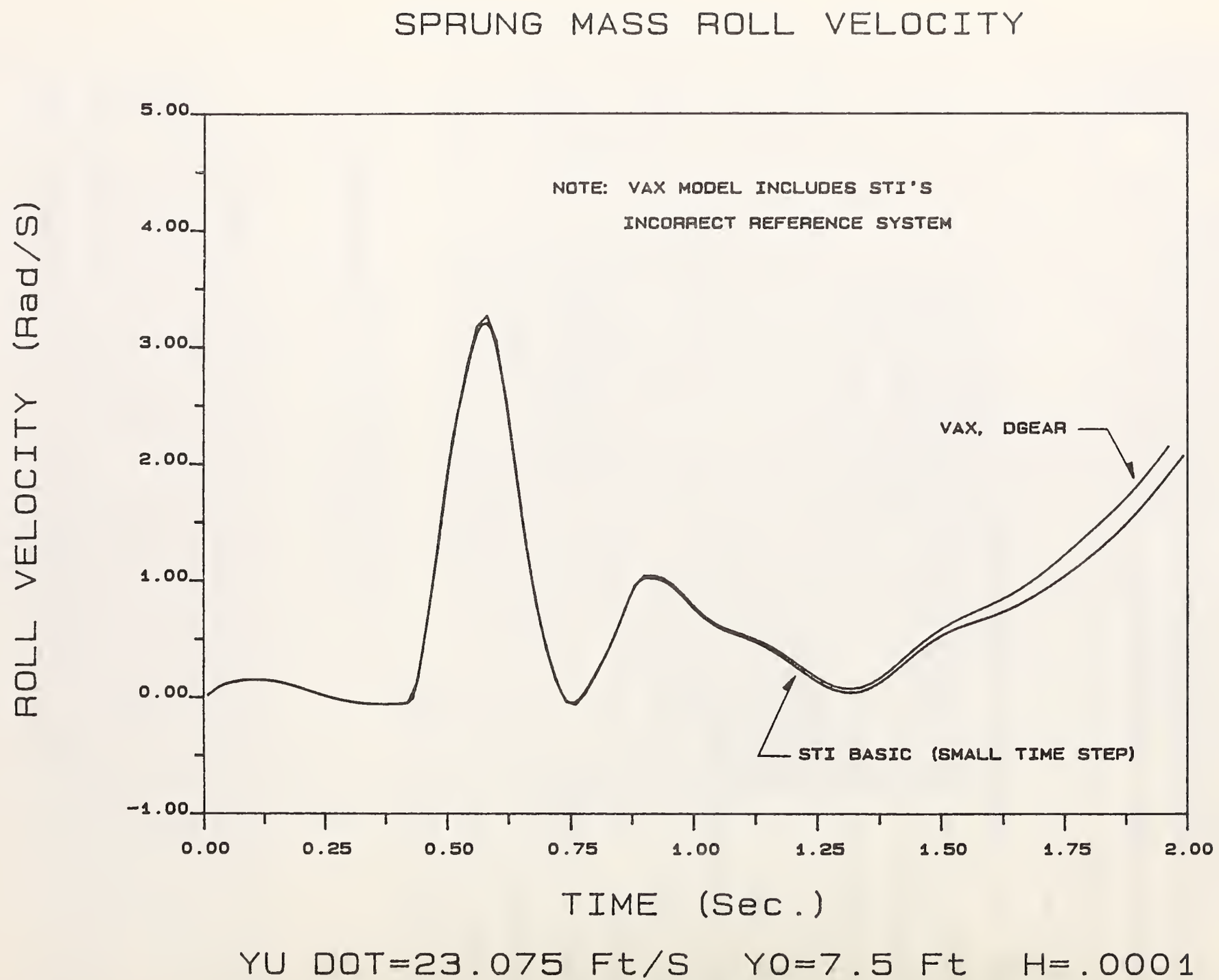

SPRUNG MASS ROLL VELOCITY
NOTE: VAX MODEL INCLUDES STI'S
INCORRECT REFERENCE SYSTEM
VAX, DGEAR
STI BASIC (SMALL TIME STEP)
ROLL VELOCITY (Rad/S)
5.00
4.00
3.00
2.00
1.00
0.00
-1.00
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
TIME (Sec.)
YU DOT=23.075 Ft/S YO=7.5 Ft H=.0001

Fig. 8

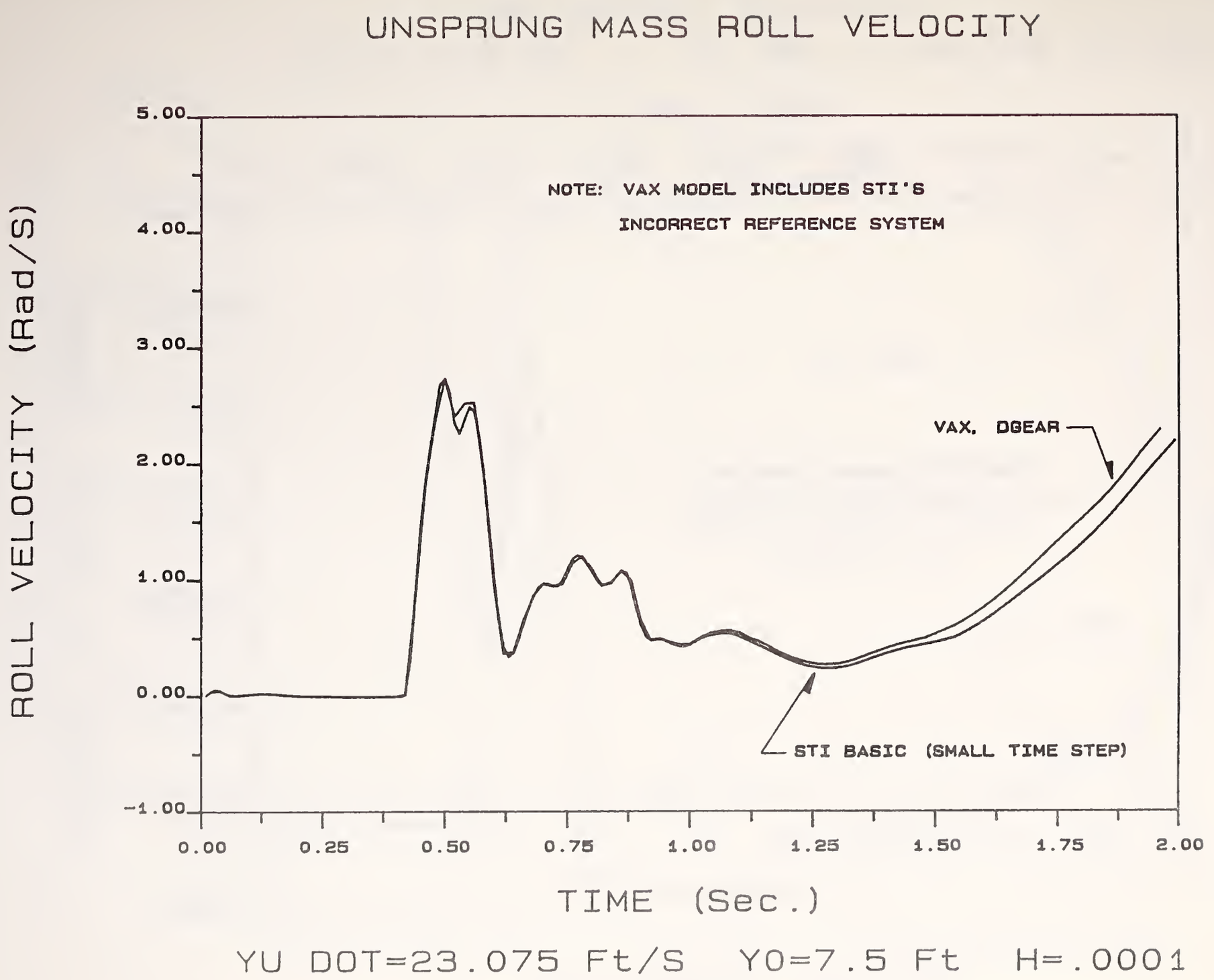

UNSPRUNG MASS ROLL VELOCITY
NOTE: VAX MODEL INCLUDES STI'S
INCORRECT REFERENCE SYSTEM
VAX, DGEAR
STI BASIC (SMALL TIME STEP)
ROLL VELOCITY (Rad/S)
5.00
4.00
3.00
2.00
1.00
0.00
-1.00
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
TIME (Sec.)
YU DOT=23.075 Ft/S YO=7.5 Ft H=.0001

Fig. 9

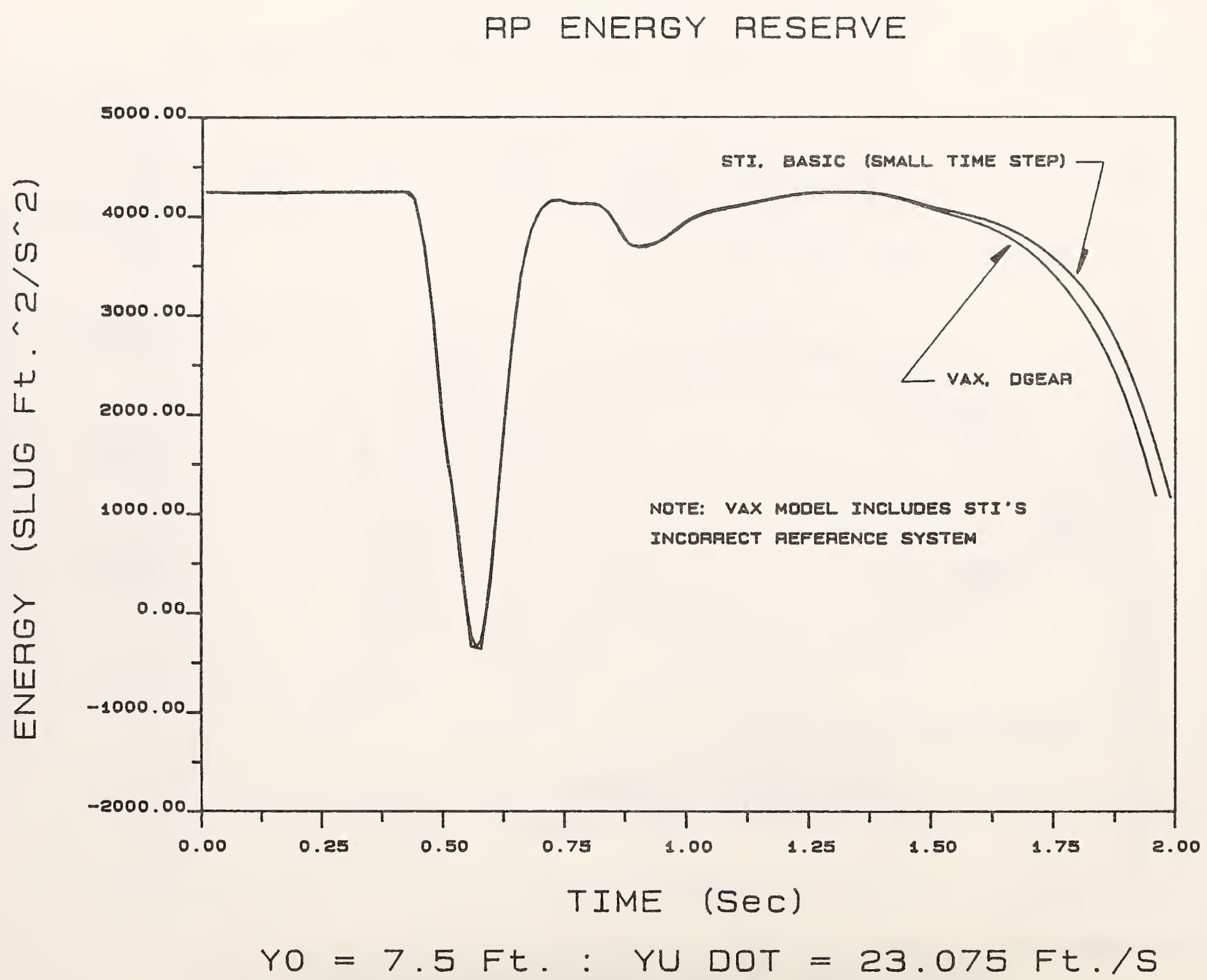

RP ENERGY RESERVE
ENERGY (SLUG Ft.^2/S^2)
5000.00
4000.00
3000.00
2000.00
1000.00
0.00
-1000.00
-2000.00
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
TIME (Sec)
STI, BASIC (SMALL TIME STEP)
VAX, DGEAR
NOTE: VAX MODEL INCLUDES STI'S
INCORRECT REFERENCE SYSTEM
Y0 = 7.5 Ft. : YU DOT = 23.075 Ft./S

Fig. 10

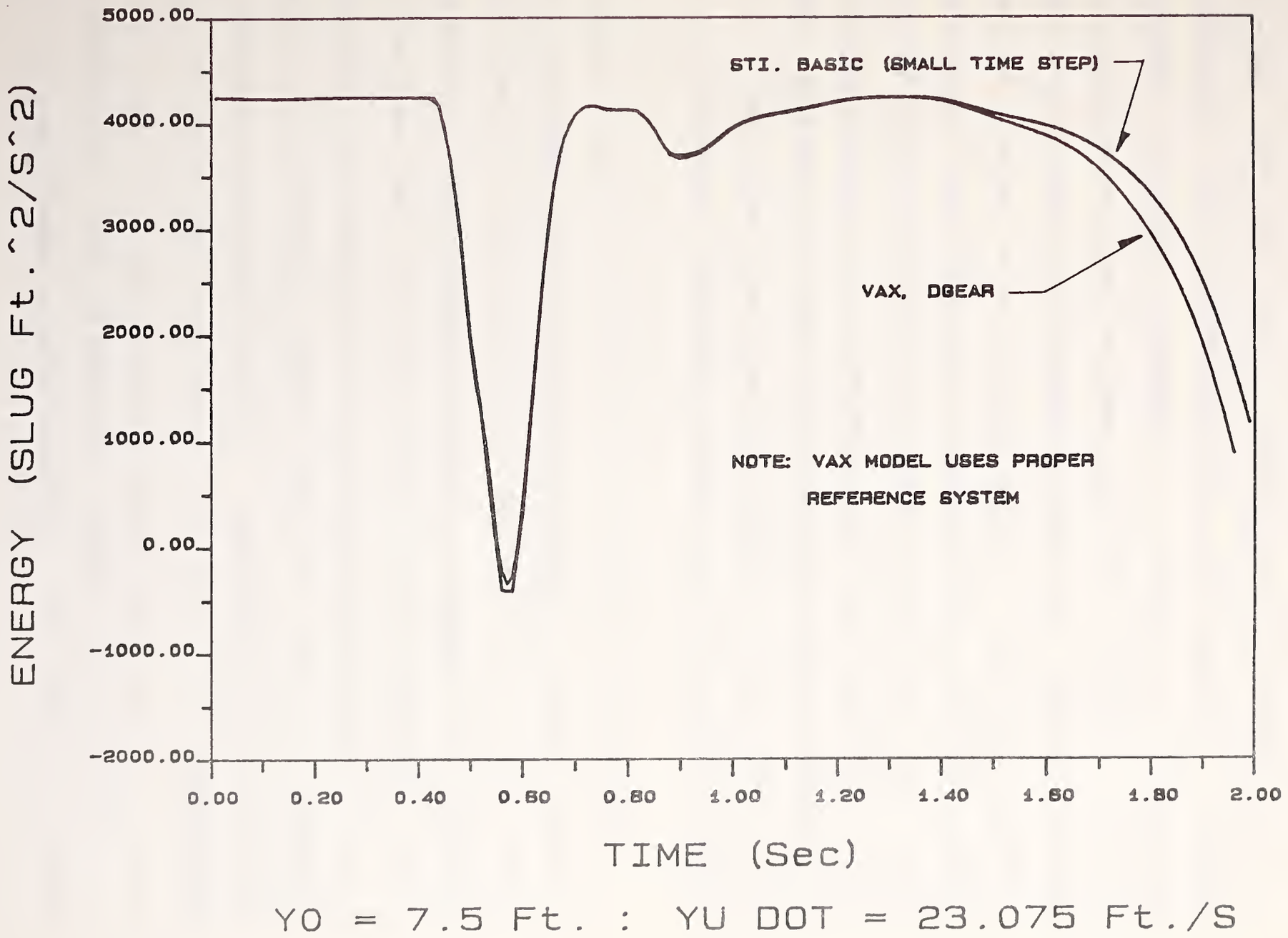

RP ENERGY RESERVE
ENERGY (SLUG Ft.^2/S^2)
5000.00
4000.00
3000.00
2000.00
1000.00
0.00
-1000.00
-2000.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
STI. BASIC (SMALL TIME STEP)
VAX, DGEAR
NOTE: VAX MODEL USES PROPER
REFERENCE SYSTEM
YO = 7.5 Ft. : YU DOT = 23.075 Ft./S

Fig. 11

4.0 SENSITIVITY METHODS EMPLOYED IN ANALYSIS

A vehicle's dynamic system and its subsystems (suspensions, brakes, steering, ect.) are comprised of many elements whose characteristics directly or indirectly affect its behavior in critical accident situations such as those encountered during vehicle rollover. An understanding of how these characteristics affect a vehicle's dynamic behavior is an essential part of design process. Unfortunately, the influence of system parameters on a vehicle's response in a particular maneuver is not always immediately evident. However, it is possible to efficiently determine the influence of all parameters on system response using sensitivity methods, making them very useful in the analysis and synthesis of complex mechanical systems.

Sensitivity analysis can be carried out in both the time and frequency domains. When the dynamic characteristics of a vehicle system are given in the form of amplitude/phase frequency characteristics or equivalent frequency transfer functions then sensitivity functions in the frequency domain are the most useful. If a vehicle/driver/environment system is given by differential equations of motion and the transient response is of interest then sensitivity analysis in the time domain is the most appropriate.

As part of the research sponsored by NHTSA the University of Missouri-Columbia has developed several general purpose algor-

ithms which can be used to calculate the sensitivity functions of mechanical systems. Complete descriptions of these sensitivity algorithms and the methods developed by UMC are given in references [7] and [8]. The algorithms are capable of analyzing virtually any system, regardless of its complexity, and permit analysis of nonlinear transient mechanical models in the time domain, nonlinear steady state vehicle models, as well as linear mechanical models in the frequency domain. The algorithms allow investigation of six different types of sensitivity functions including first order standard, first order percentage, first order logarithmic, second order standard, the logarithmic general sensitivity measure, and the percentage general sensitivity measure. Deciding which type of sensitivity functions should be used depends on the objectives of the user. A brief description of each type of sensitivity function is given below.

4.1 First Order Standard Sensitivity Function

The first order standard sensitivity function represents the partial derivative of a system variable taken with respect to a particular system parameter. This type of sensitivity function is useful for determining the influence of a particular parameter on a particular system variable. For example, if a designer wished to investigate the influence of wheelbase on vehicle yaw velocity then first order standard sensitivity functions could be used.

4.2 First Order Percentage Sensitivity Function

The first order percentage sensitivity function represents the variable change caused by a parameter change of some user specified percentage. Percentage sensitivity functions are useful for comparing the influence of various parameters on a particular system variable. For example, if a designer wanted to know if a 2 percent change in wheelbase effected yaw angular velocity more than a 2 percent change in track width the percentage sensitivity function would show the change in yaw velocity associated with each parameter change. Because percentage sensitivity functions have units which are identical to that of the system variable, they can be used to compare the effects of various parameters on a particular variable.

4.3 First Order Logarithmic Sensitivity Function

The first order logarithmic sensitivity function is the only dimensionless sensitivity function available, making it useful to compare and rank the influence of various parameters on system variables. Users should be aware that if the variable value approaches zero then the logarithmic sensitivity functions can become very large. It should also be noted that logarithmic sensitivity is dependent on both system sensitivity and system response, which can sometimes make interpretation of the time behavior of a logarithmic sensitivity function difficult. However, logarithmic sensitivity functions are useful in comparing

the order of influence of various system parameters at a particular time.

4.4 Second Order Standard Sensitivity Function

Second order sensitivity functions represent the second partial derivative of a system variable taken with respect to a system parameter. The second order sensitivity function can be used to gain additional insight into results obtained using first order standard sensitivity functions. When the system exhibits similar sensitivity to a number of parameters then second order sensitivity functions are particularly useful.

4.5 Logarithmic and Percentage General Sensitivity Measures

The general sensitivity measure determines the global effect of all selected parameters on a particular variable. This is useful for analyzing the combined effects of related parameters. For example, one could compare the combined effects of parameters describing the system's stiffness with parameters describing the system's mass properties. The general sensitivity measure can be based on either percentage sensitivity or logarithmic sensitivity functions since these types possess compatible dimensions.

The different types of sensitivity functions can be used when carrying out sensitivity analysis. For certain situations the most recommended sensitivity functions are given below.

Type of Sensitivity Function	Application
First order standard	- one parameter on one variable
Percentage and logarithmic	- many parameters on one variable
First order logarithmic	- one parameter on several variables
General sens. measure (perc.)	- various groups of parameters (geom., mass, stiff., ect.) on one variable
General sens. measure (log.)	- global effect of many parameters (in groups) on several variables
Second order standard	- when system exhibits similar sensitivity to a number of parameters

4.6 Numerical Methods Used in Sensitivity Analysis

Because the main objective of this project was to perform a sensitivity analysis on STI's tripped rollover simulation and compare the importance of system parameters on rollover behavior it was decided that percentage sensitivity functions best suited the purposes of this project. Logarithmic sensitivity was deemed inappropriate since its dependence on system response could complicate the analysis of sensitivity results.

Time domain sensitivity functions are computed by integrating the mechanical system's differential equations of motion using different sets of parameter values. The equations of motion are first integrated using nominal parameter values. The

parameter of interest is then increased by some user specified percentage (usually 1%) and the equations of motion are integrated again to obtain a varied system response. The nominal parameter value is then reduced by the user specified percentage and the equations of motion are integrated again to obtain a second varied system response. Once the system response has been determined using the different sets of parameter values then the first order standard sensitivity and second order standard sensitivity functions can be computed by taking the first and second partial derivatives of the variable of interest with respect to the parameter using a three point central difference method. The sensitivity algorithms are also capable of using either 5 or 7 point central difference method to compute the partial derivatives, however the sensitivity algorithm must carry out additional model integrations to use these methods. First order logarithmic sensitivity is computed by multiplying the first order standard sensitivity function by the nominal parameter value and then dividing it by the variable value obtained using the nominal parameter set. Percentage sensitivity is computed by finding the difference in the variable caused by the user specified parameter change. Table 1 shows the differences between the various sensitivity methods:

Table 1.

Sensitivity Type	Analytical Expression	Numerical Expression	Unit
1st Order Standard	$\frac{\partial V}{\partial P}$	$\frac{\Delta V}{\Delta P}$	$\frac{V}{P}$
1st Order Percentage	$\frac{\delta V}{\delta P}\ \delta P$	ΔV	V
1st Order Logarithmic	$\frac{\partial V}{\partial P}\ \frac{P}{V}$	$\frac{\Delta V}{\Delta P}\ \frac{P}{V}$	None
2nd Order Standard	$\frac{\partial^2 V}{\partial P^2}$	$\frac{\Delta\left(\frac{\Delta V}{\Delta P}\right)}{\Delta P}$	$\frac{V}{P^2}$
General Sensitivity Measure	$GS_V = \left[\sum_{i=1}^{n} S_{iv}^2\right]^{\frac{1}{2}}$	(Where S_{iv} is any of the sensitivity functions shown above)	

In the case of secondary parameters (parameters whose values do not remain constant e.g. tire forces) the central difference expressions cannot be used since they assume that the distance between varied parameter values is constant. Therefore under these conditions the sensitivity algorithms use a polynomial fit to determine the first and second order standard sensitivity functions. The program uses a 2nd order polynomial when using the three point method, a 4th order polynomial for the five point method and a 6th order polynomial for the seven point method. A more complete description of the methods used in the sensitivity algorithms is given in references [7] and [8].

5.0 ENERGY ANALYSIS OF STI TRIPPED ROLLOVER MODEL

Vehicle rollover is essentially a problem associated with stability. In a rollover situation, a vehicle which generates small roll angles will return to a four wheel stance and is considered stable, however at larger roll angles the vehicle can tip over and is considered unstable. The stability of a system's equilibrium position can be investigated based on the change in the system's potential energy caused by a small perturbation of its state variables. If this change is positive then the system is stable, if negative then the system is unstable. For example, when a vehicle has a roll angle of zero an increase in the vehicle roll angle raises its center of gravity position, thus increasing the potential energy of the system, which indicates that the initial position of a vehicle is stable. However, when a vehicle's roll angle equals the static tip over angle then increasing the roll angle would lower the c.g. position, decreasing the potential energy of the system, making the vehicle unstable. While it was deemed impractical to perform a classical stability analysis on STI's Tripped Rollover Vehicle Simulation, it was possible to examine the kinetic and potential energies of the vehicle. This was done in order to gain insight into the development of a function which could be used as an indication of vehicle rollover propensity. It was believed that the maximum allowable change in the gravitational potential energy from the static condition to the static tip over position might serve as a threshold which the non-centroidal rotational kinetic energy must surpass in order for vehicle rollover to occur.

For the sets of initial conditions considered in this report, vehicle yawing and pitching angular velocities were small and the kinetic energies associated with these motions were neglected. Under this assumption the kinetic energy of general plane motion (T_{GPM}) represents the total kinetic energy of the vehicle and is equal to the sum of the system's kinetic energies due to the sprung and unsprung masses' c.g. velocities and rolling angular velocities:

$$T_{GPM} = \frac{1}{2} M_s V_s^2 + \frac{1}{2} M_u V_u^2 + \frac{1}{2} I_s \omega_s^2 + \frac{1}{2} I_u \omega_u^2$$

The kinetic energy of the general plane motion was divided into two parts. The kinetic energy of non-centroidal rotation (T_{NR}) is created during vehicle impact with the curb and represents the vehicle's kinetic energy in its rotational motion around the corner point of the curb:

$$T_{NR} = \frac{1}{2} I_{so} \omega_s^2 + \frac{1}{2} I_{uo} \omega_u^2$$

The kinetic energy of translation (T_{TRAN}) is the equal to the kinetic energy of general plane motion minus the kinetic energy of non-centroidal rotation:

$$T_{TRAN} = T_{GPM} - T_{NR}$$

The gravitational potential energy of the vehicle (V_G) was calculated relative to its static equilibrium position:

$$V_G = (M_s + M_u) \cdot g \cdot (h_3 - h_{C.G.})$$

where: $h_{C.G.}$ = static average C.G. Height

h_3 = instantaneous average C.G. Height

Thus, for a vehicle to attain a maximum value of gravitational potential energy the vehicle's center of gravity must be placed at the highest possible point (the static tipover angle).

The change in elastic potential energy stored in the suspension springs and the vehicle tires was investigated, however it was difficult to determine what portion of this energy was a contributing factor to rollover and what portion was dissipated. In the final analysis, the change in elastic potential energy was not included.

5.1 Rollover Prevention Energy Reserve (RPER)

It was felt that in order for a vehicle to rollover, the kinetic energy of non-centroidal rotation (T_{NR}), created during the vehicle's impact with the curb, must exceed the maximum allowable change in gravitational potential energy (V_{CRIT}) of the vehicle. A vehicle's rollover prevention energy reserve (RPER) was then developed using these two quantities as an indicator of

rollover stability. The rollover prevention energy reserve is defined:

$$RPER = V_{CRIT} - T_{NR}$$

$$= (M_s + M_u) \cdot g \cdot \left[\sqrt{h_{C.G.}^2 + \left(\frac{TRW}{2}\right)^2} - h_{C.G.}\right] - \frac{1}{2} I_{so} \omega_s^2 + \frac{1}{2} I_{uo} \omega_u^2$$

$$= (M_s + M_u) \cdot g \cdot h_{C.G.} \cdot \left[\sqrt{1 + SRSF^2} - 1\right] - \frac{1}{2} I_{so} \omega_s^2 + \frac{1}{2} I_{uo} \omega_u^2$$

where : $$SRSF = \frac{TRW}{2 \cdot h_{C.G.}}$$

The kinetic energy of the vehicle is converted into gravitational potential energy when the vehicle rolls. A vehicle which strikes the curb at the critical rollover side velocity (the speed at which the the vehicle barely rolls over) should have almost no kinetic energy at the static rollover angle (the point of maximum gravitational potential energy). Also, due to energy dissipation, the vehicle's kinetic energy of non-centroidal rotation must exceed its maximum gravitational potential energy by the amount of which is energy dissipated by tire, suspension and impact forces after the instant when T_{NR} reaches its maximum value and before the static rollover angle has been reached.

5.2 Results of Energy Analysis

The energy analysis of the tripped rollover model was conducted at various speeds in order to study the results in rollover and non-rollover cases. The time histories of various energy components were examined (Fig. 12 - 16) using two sets of initial conditions in which the vehicle did not rollover (initial side velocity equal to 17 Ft/S and 22 Ft/S) , as well as three sets which produced vehicle rollover (23.075 Ft/S, 25 Ft/S, and 27 Ft/S). The initial distance of the front wheel to the curb was equal to 7.5 Ft. and the initial heading angle of the vehicle was equal to zero (resulting in simultaneous impact of front and rear wheels) in each of the five cases. The lateral velocity of 23.075 Ft/S was designated as the critical rollover side velocity. The results show that, after curb impact, the kinetic energy of non-centroidal rotation increases to values which are greater than the total kinetic energy of general plane motion, resulting in values of the translational kinetic energy which are less than zero. The absence of several dynamic couplings in the equations of motion in STI's vehicle model is believed to be the cause of this discrepancy.

The rollover prevention energy reserve graphs for each of the five different test speeds are shown in Fig. 17 - 21. With a lateral velocity of 17 Ft/S, the vehicle came to a complete stop prior to impact with the curb. Figure 17 shows a slight reduction in the vehicle's RPER between .8 second and 1 second due to

the change in chassis roll angle which returns to its static position when the vehicle comes to rest.

At the lateral velocity of 22 Ft/S (Fig. 18) the vehicle did impact the curb, but did not rollover. The vehicle impacted the curb, rolled to a maximum roll angle smaller than the critical roll angle (the static tip over angle of the vehicle), and then returned to a stable position on four wheels. The chassis continued to roll after returning to a four wheel stance, causing rolling oscillations which were damped out by the shock absorbers. Figure 18 shows a significant drop in the RPER immediately after curb impact, followed by oscillations which were damped out as the vehicle came to a resting position. In this and all other non-rollover cases examined, the RPER remained positive throughout the simulation.

The first rollover case examined was at the critical rollover lateral velocity (23.075 Ft/S, Fig. 19). Immediately after curb impact, the vehicle's RPER dropped sharply to a value of approximately -400 Slug Ft^2/S^2. The RPER then increased as the vehicle's roll angle approached the static tip over value. At this point the roll angular velocity of the vehicle was nearly zero. The RPER then drops again once the vehicle passes through the static tip over position and rolls over.

The next two cases examine the RPER using initial lateral velocities of 25 Ft/S and 27 Ft/S (Fig. 20 and 21). The figures show that the higher impact speeds result in larger drops in

vehicle RPER. At an initial speed of 25 Ft/S the vehicle's RPER dropped to a value of -10000 Slug Ft^2/ S^2 immediately after impact. The vehicle's RPER then increases to nearly zero, but then falls again as the vehicle rolls over. At an initial speed of 27 Ft/S the decrease in vehicle RPER is much greater (Fig. 21). Immediately after impact it's RPER drops to approximately -25000 Slug Ft^2/ S^2 and then increases to -16000 Slug Ft^2 / S^2 before falling again as the vehicle rolls over.

Figure 22 shows vehicle RPER at each of the five initial speeds. The figure clearly illustrates that the minimum value of RPER is positive at speeds which are smaller than the critical rollover velocity and negative at speeds greater than the critical rollover velocity, which confirms that RPER is a very useful indicator of vehicle rollover propensity in tripped rollover maneuvers with simultaneous front and rear wheel impact.

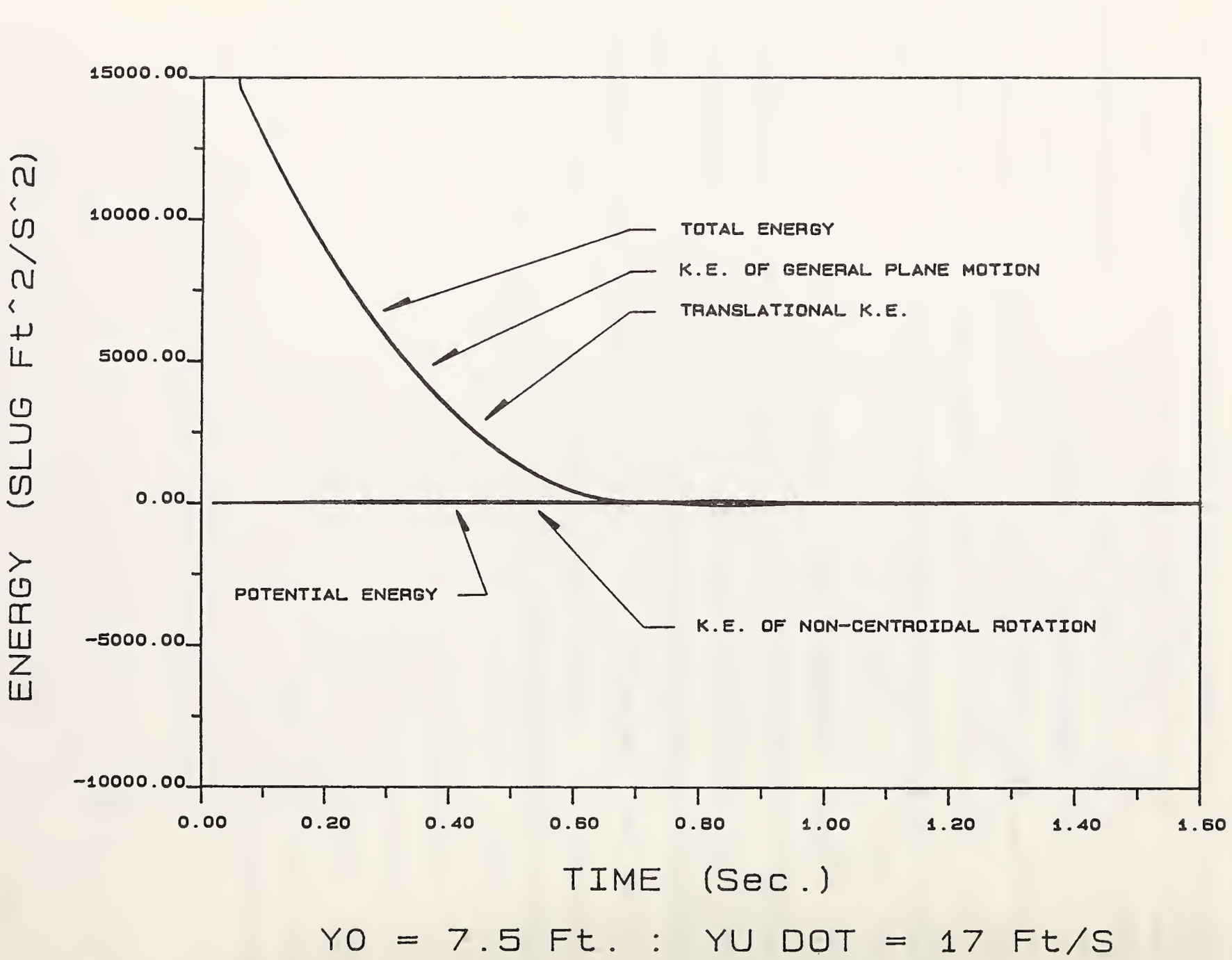

ENERGY ANALYSIS OF STI MODEL
15000.00
10000.00
5000.00
0.00
-5000.00
-10000.00
ENERGY (SLUG Ft^2/S^2)
TOTAL ENERGY
K.E. OF GENERAL PLANE MOTION
TRANSLATIONAL K.E.
POTENTIAL ENERGY
K.E. OF NON-CENTROIDAL ROTATION
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
TIME (Sec.)
YO = 7.5 Ft. : YU DOT = 17 Ft/S

Fig. 12

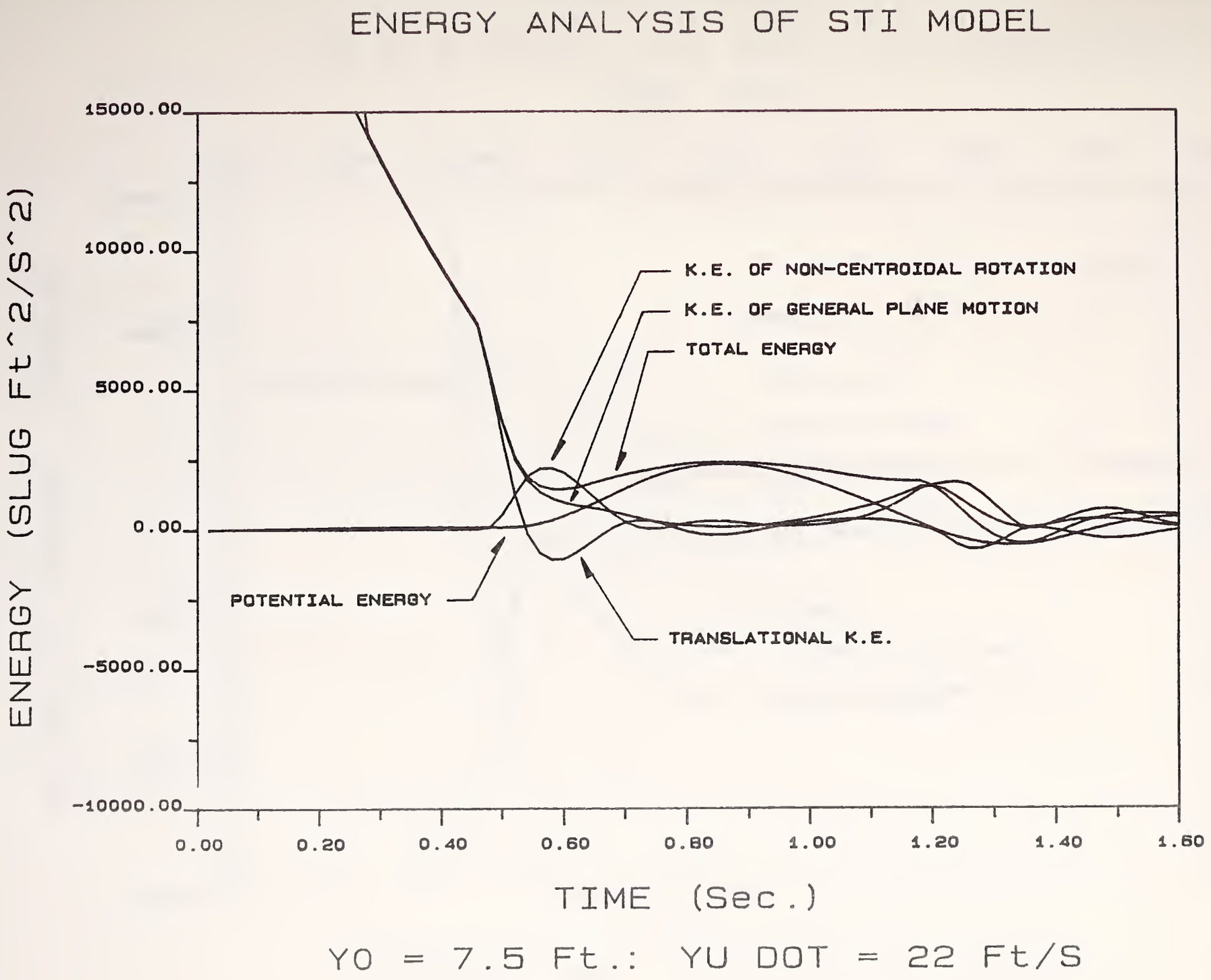
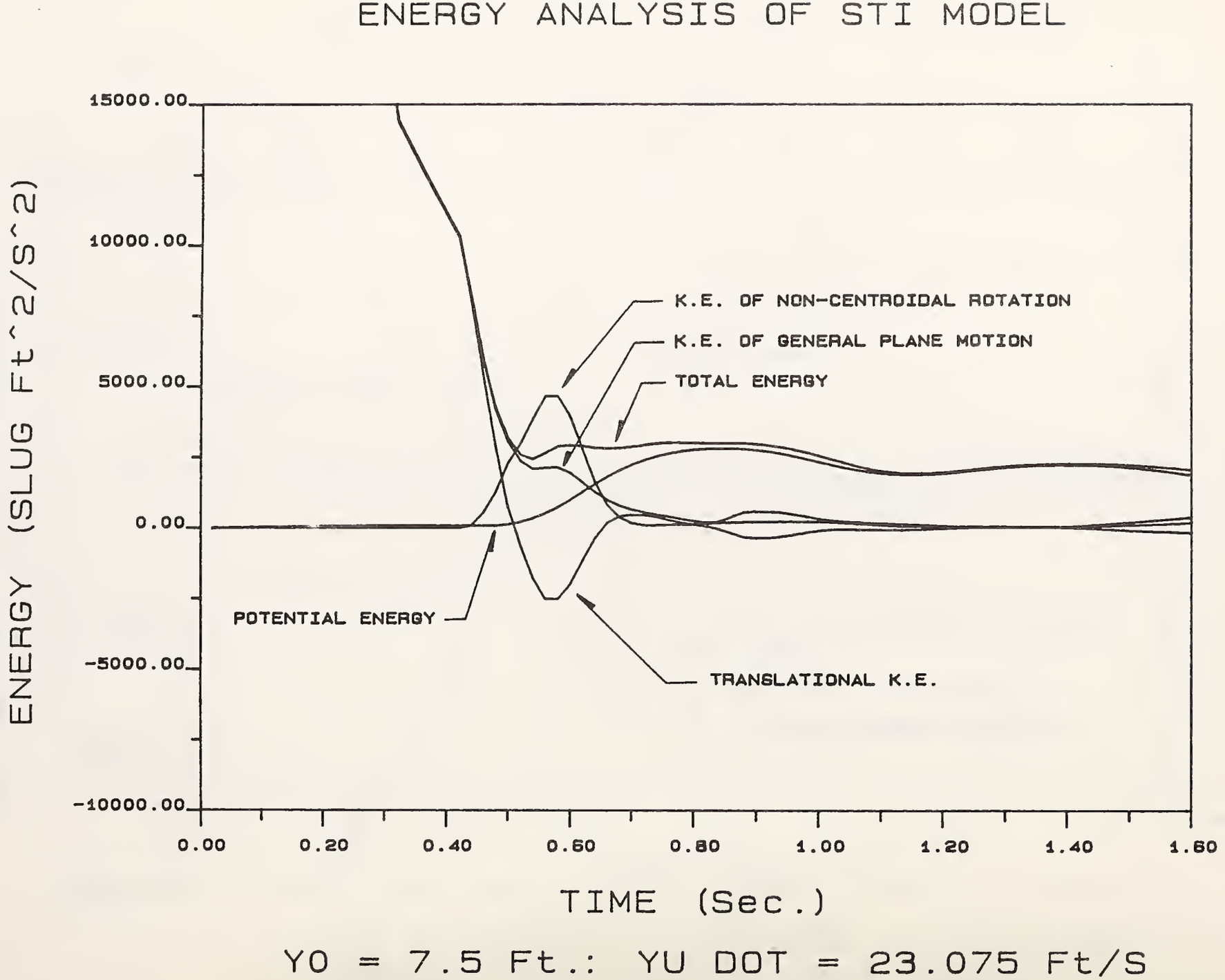

ENERGY ANALYSIS OF STI MODEL
15000.00
10000.00
5000.00
0.00
-5000.00
-10000.00
ENERGY (SLUG Ft^2/S^2)
K.E. OF NON-CENTROIDAL ROTATION
K.E. OF GENERAL PLANE MOTION
TOTAL ENERGY
POTENTIAL ENERGY
TRANSLATIONAL K.E.
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
TIME (Sec.)
YO = 7.5 Ft.: YU DOT = 22 Ft/S

Fig. 13


ENERGY ANALYSIS OF STI MODEL
ENERGY (SLUG Ft^2/S^2)
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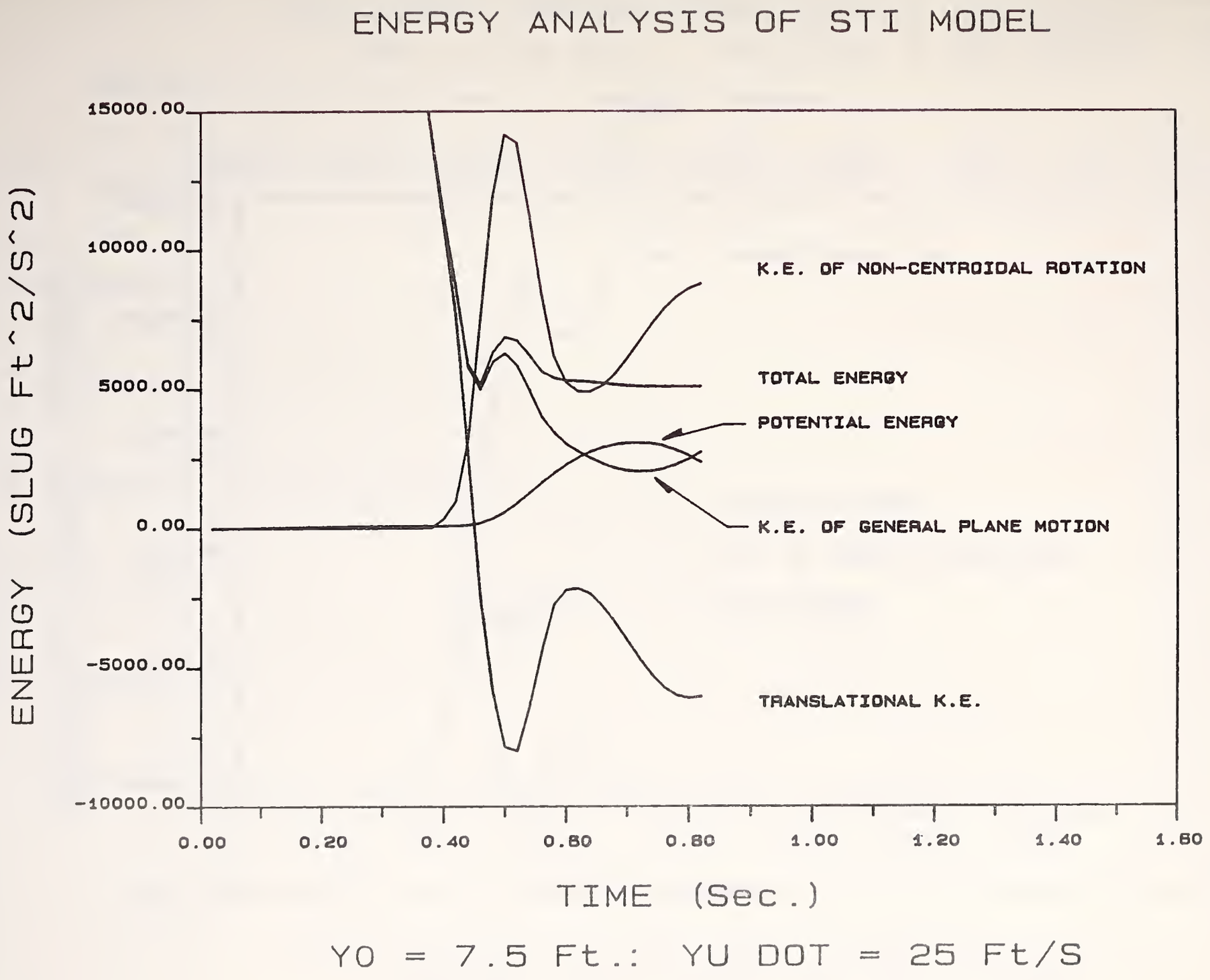
10000.00
5000.00
0.00
-5000.00
-10000.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
TIME (Sec.)
K.E. OF NON-CENTROIDAL ROTATION
K.E. OF GENERAL PLANE MOTION
TOTAL ENERGY
POTENTIAL ENERGY
TRANSLATIONAL K.E.
YO = 7.5 Ft.: YU DOT = 23.075 Ft/S

Fig. 14


ENERGY ANALYSIS OF STI MODEL
ENERGY (SLUG Ft^2/S^2)
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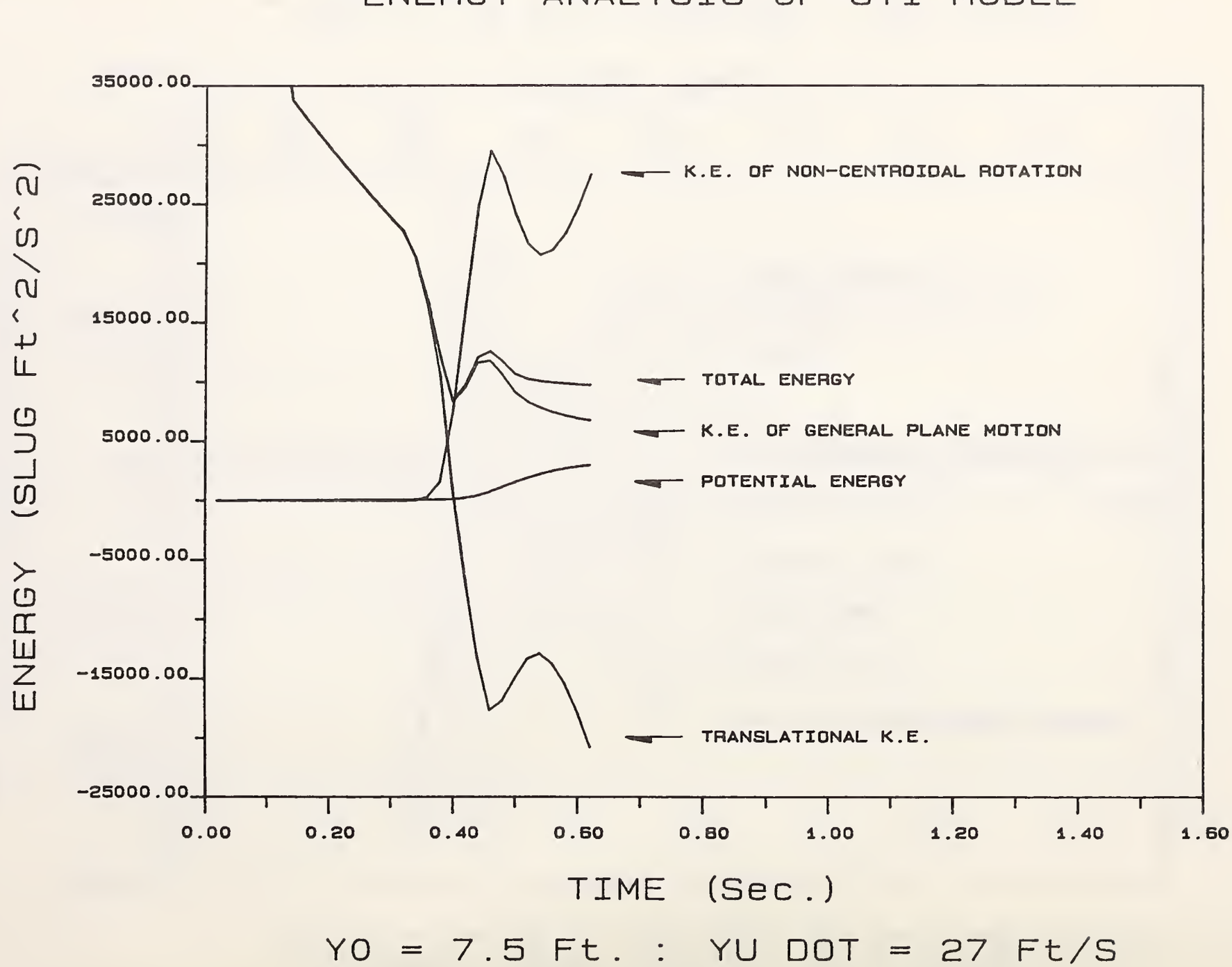
10000.00
5000.00
0.00
-5000.00
-10000.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
TIME (Sec.)
K.E. OF NON-CENTROIDAL ROTATION
TOTAL ENERGY
POTENTIAL ENERGY
K.E. OF GENERAL PLANE MOTION
TRANSLATIONAL K.E.
YO = 7.5 Ft.: YU DOT = 25 Ft/S

Fig. 15


ENERGY ANALYSIS OF STI MODEL
ENERGY (SLUG Ft^2/S^2)
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25000.00
15000.00
5000.00
-5000.00
-15000.00
-25000.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
TIME (Sec.)
K.E. OF NON-CENTROIDAL ROTATION
TOTAL ENERGY
K.E. OF GENERAL PLANE MOTION
POTENTIAL ENERGY
TRANSLATIONAL K.E.
YO = 7.5 Ft. : YU DOT = 27 Ft/S

Fig. 16

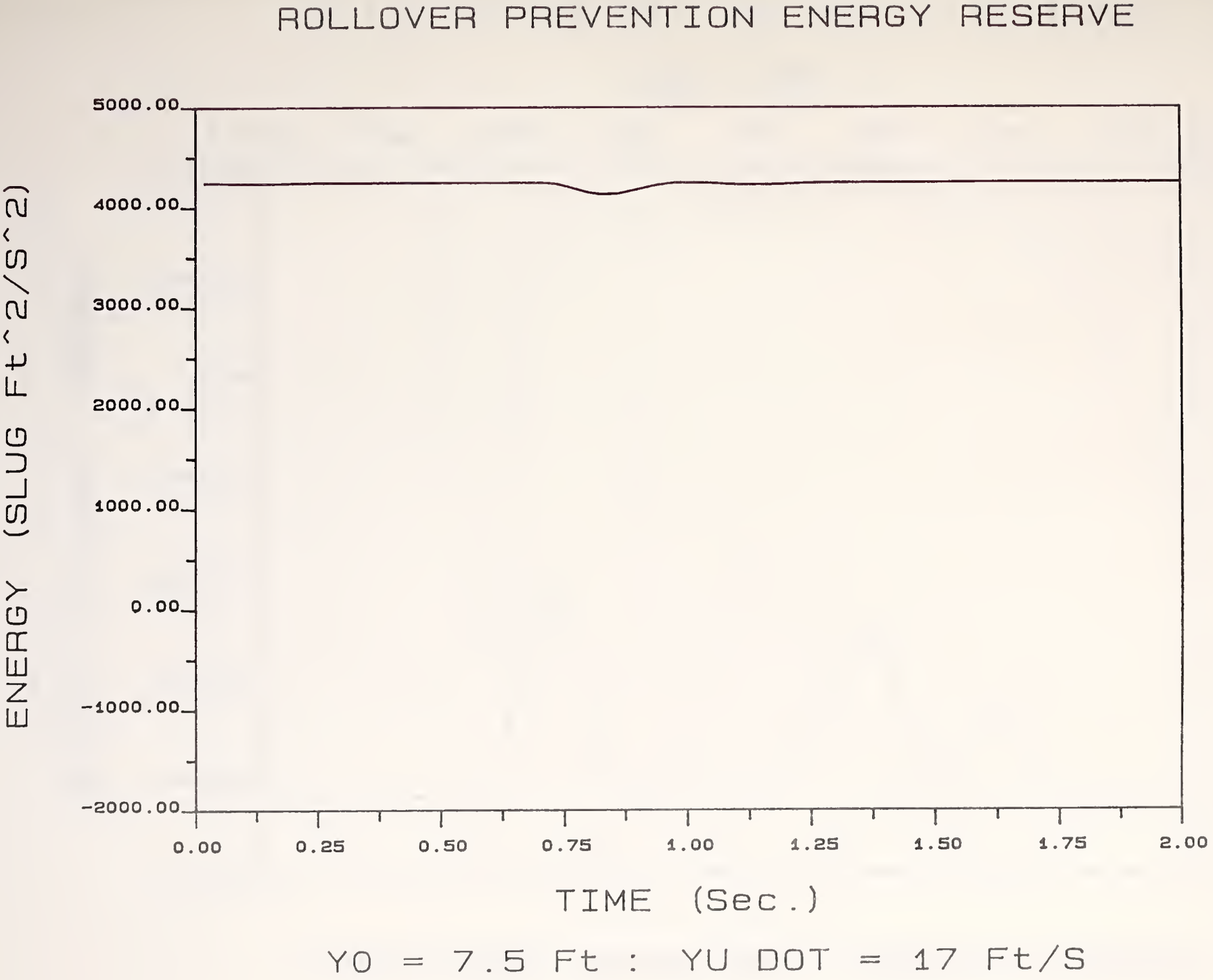

ROLLOVER PREVENTION ENERGY RESERVE
ENERGY (SLUG Ft^2/S^2)
5000.00
4000.00
3000.00
2000.00
1000.00
0.00
-1000.00
-2000.00
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
TIME (Sec.)
YO = 7.5 Ft : YU DOT = 17 Ft/S

Fig. 17

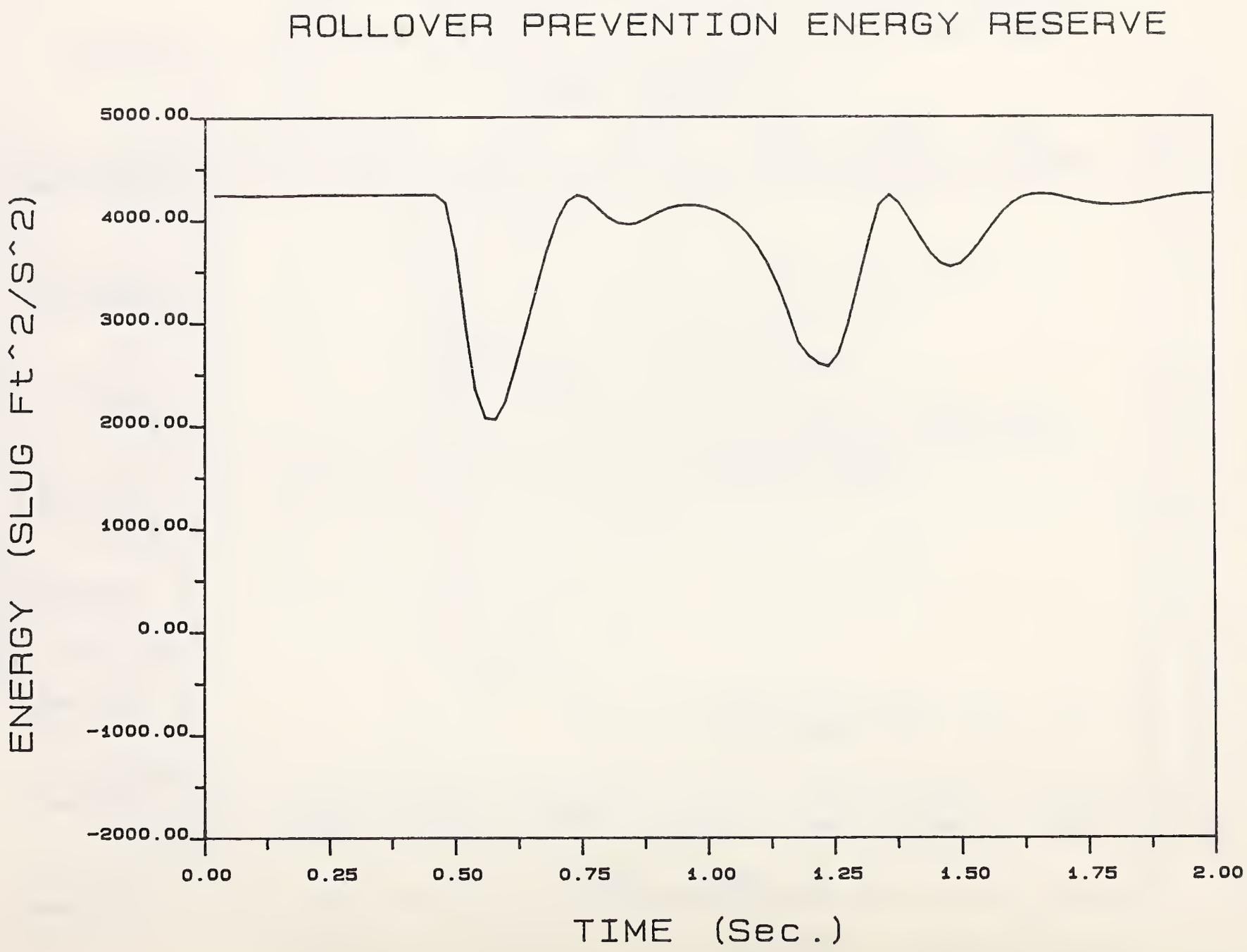

ROLLOVER PREVENTION ENERGY RESERVE
ENERGY (SLUG Ft^2/S^2)
5000.00
4000.00
3000.00
2000.00
1000.00
0.00
-1000.00
-2000.00
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
TIME (Sec.)
YO = 7.5 Ft : YU DOT = 22 Ft/S

Fig. 18

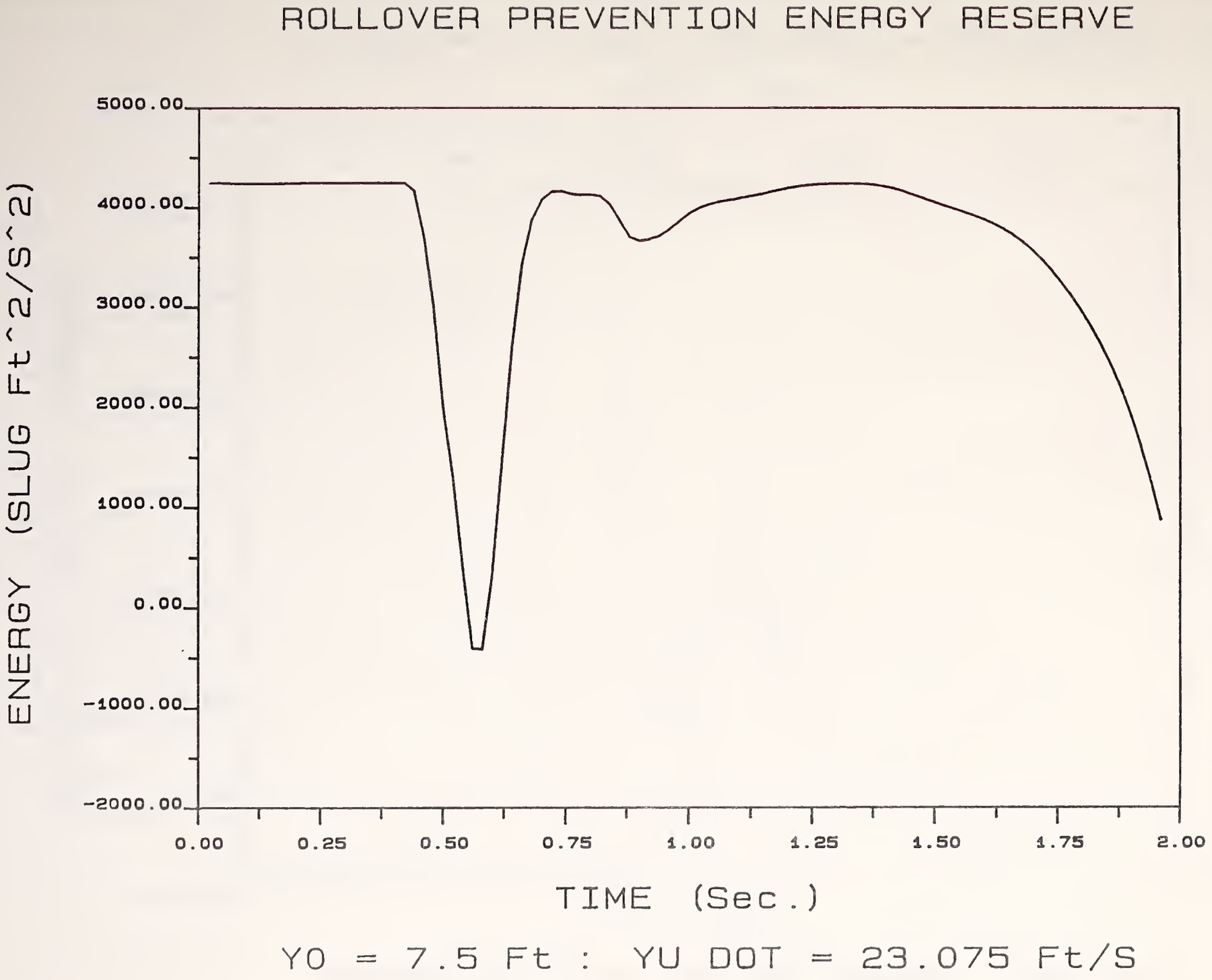

ROLLOVER PREVENTION ENERGY RESERVE
ENERGY (SLUG Ft^2/S^2)
5000.00
4000.00
3000.00
2000.00
1000.00
0.00
-1000.00
-2000.00
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
TIME (Sec.)
YO = 7.5 Ft : YU DOT = 23.075 Ft/S

Fig. 19

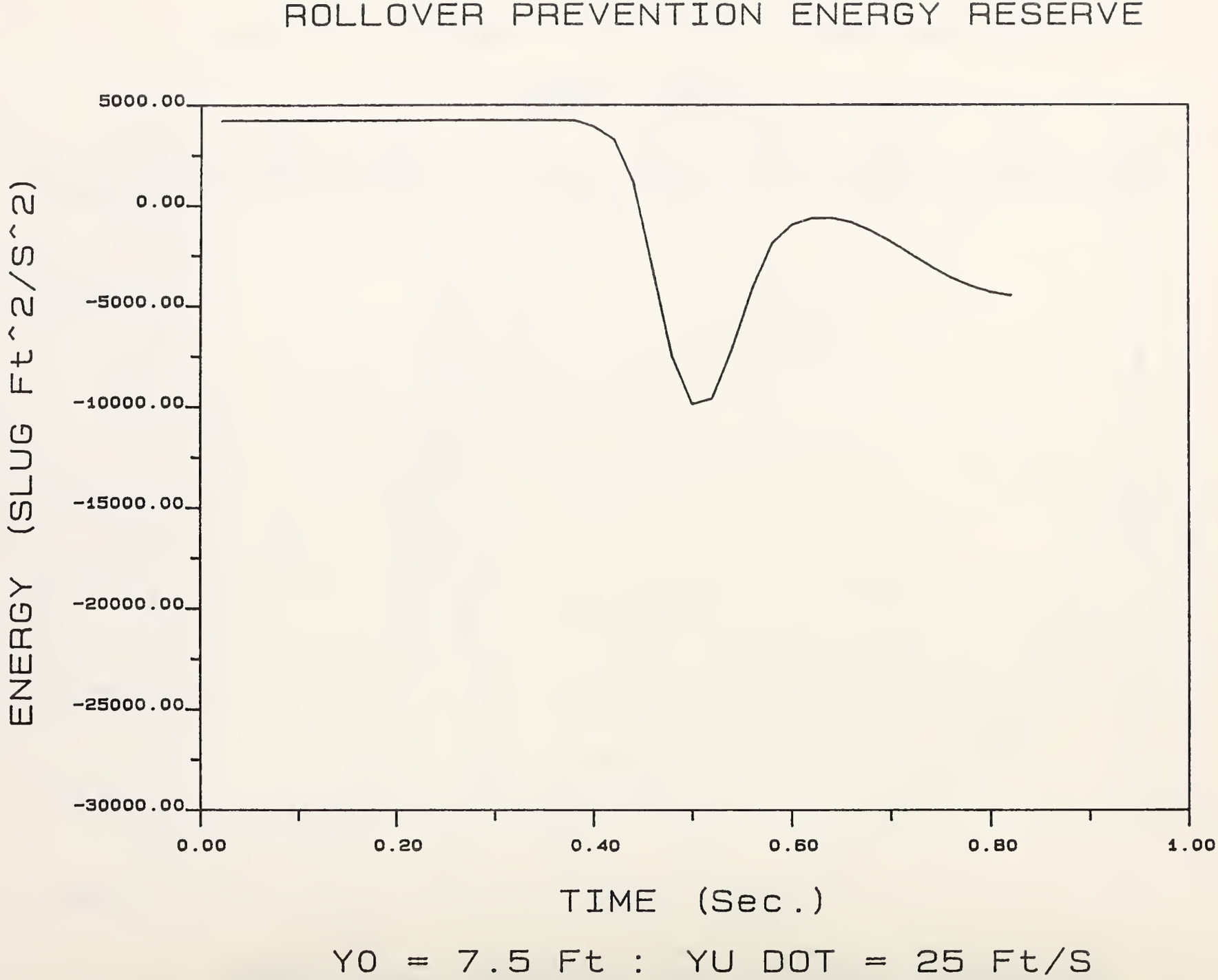

ROLLOVER PREVENTION ENERGY RESERVE
ENERGY (SLUG Ft^2/S^2)
5000.00
0.00
-5000.00
-10000.00
-15000.00
-20000.00
-25000.00
-30000.00
0.00
0.20
0.40
0.60
0.80
1.00
TIME (Sec.)
YO = 7.5 Ft : YU DOT = 25 Ft/S

Fig. 20

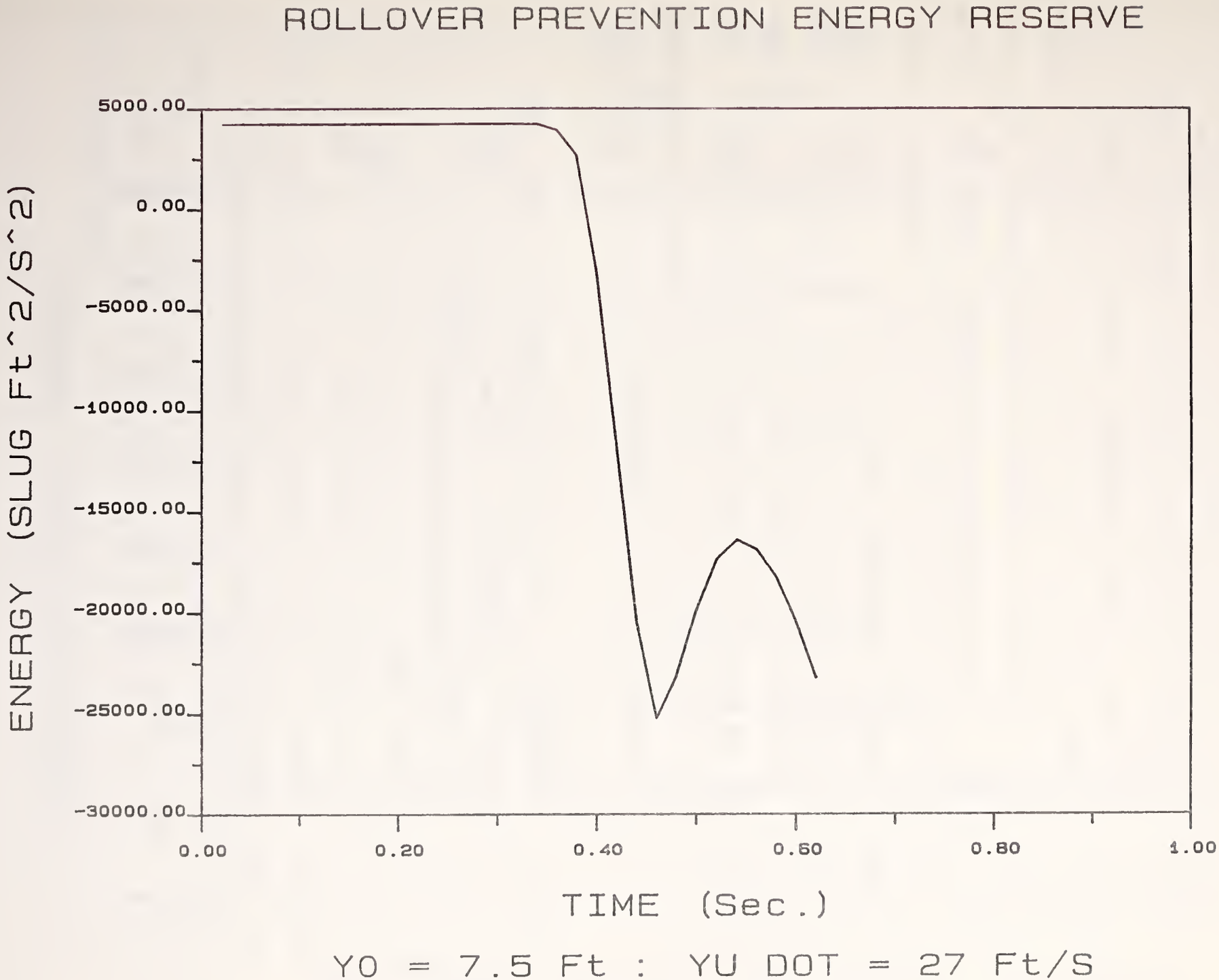

ROLLOVER PREVENTION ENERGY RESERVE
ENERGY (SLUG Ft^2/S^2)
5000.00
0.00
-5000.00
-10000.00
-15000.00
-20000.00
-25000.00
-30000.00
0.00
0.20
0.40
0.60
0.80
1.00
TIME (Sec.)
YO = 7.5 Ft : YU DOT = 27 Ft/S

Fig. 21

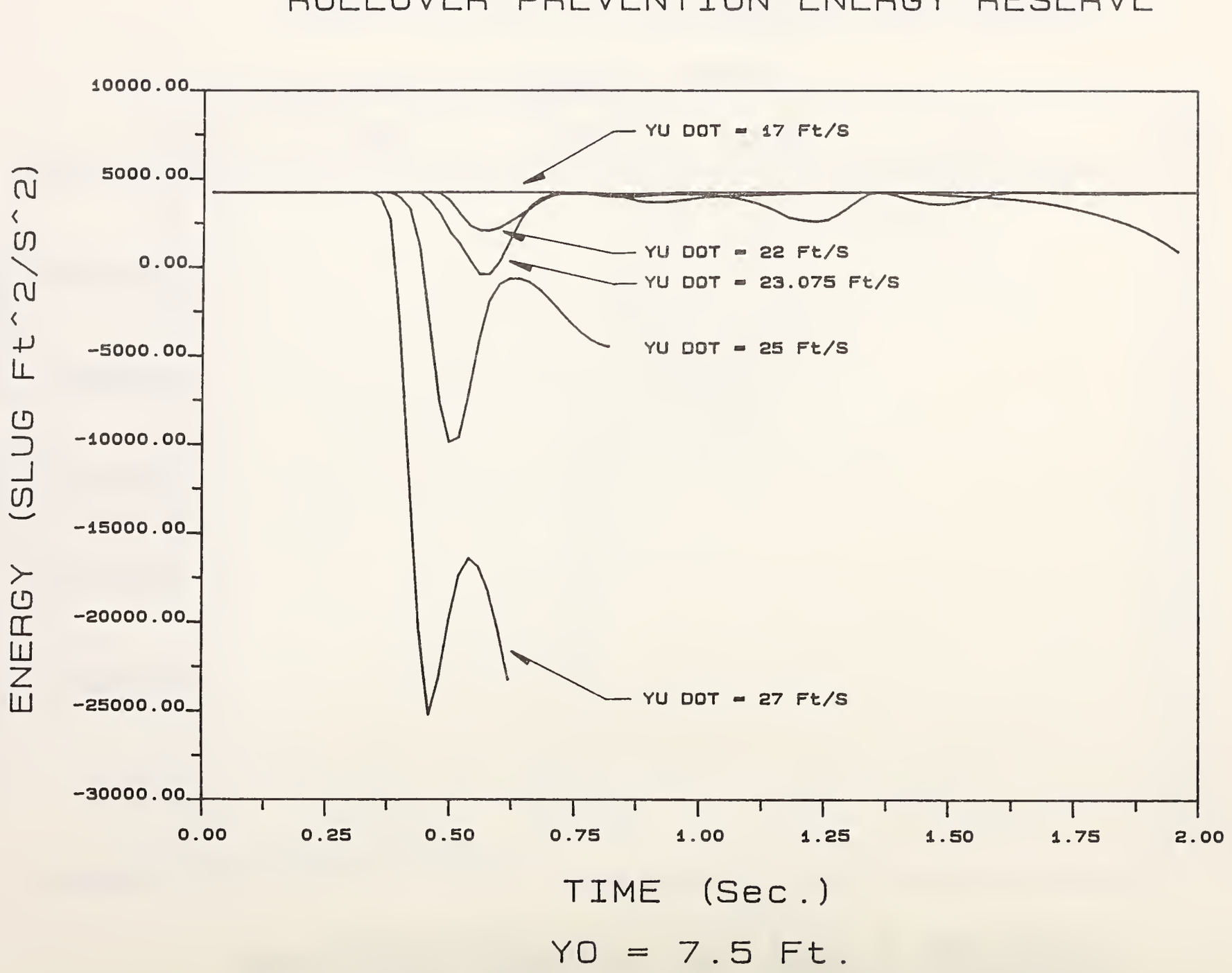

ROLLOVER PREVENTION ENERGY RESERVE
ENERGY (SLUG Ft^2/S^2)
10000.00
5000.00
0.00
-5000.00
-10000.00
-15000.00
-20000.00
-25000.00
-30000.00
YU DOT = 17 Ft/S
YU DOT = 22 Ft/S
YU DOT = 23.075 Ft/S
YU DOT = 25 Ft/S
YU DOT = 27 Ft/S
0.00
0.25
0.50
0.75
1.00
1.25
1.50
1.75
2.00
TIME (Sec.)
YO = 7.5 Ft.

Fig. 22

6.0 SENSITIVITY RESULTS

The percentage sensitivity functions of RPER were investigated for each primary (constant) vehicle parameter. All 35 vehicle parameters were split into 6 groups in order to facilitate the interpretation of results. The first two parameter groups represent geometrical parameters such as vehicle track width, wheelbase, distance from the axle to unsprung mass c.g. position, ect. The third group represents the vehicle mass and inertia parameters, the fourth stiffness parameters, the fifth damping, and the sixth group consists of parameters associated with forces such as the coefficient of friction, force of plastic suspension deformation, ect. A list and description of the parameters contained in each group is shown in Appendix E.

The percentage sensitivity functions of RPER were computed for primary parameters using initial lateral velocities of 22, 23.075 and 25 Ft/S and are shown in Figures 23 - 40. In all of these cases the initial distance from the front wheel to the curb was equal to 7.5 Ft. and the initial heading angle was zero. Vehicle impact with the curb occurs 0.3 and 0.4 seconds depending on the initial lateral velocity. It should be noted that the results obtained using the lateral speed of 25 Ft/S are plotted on a scale which is different than that used for the 22 and 23.075 Ft/S cases.

6.1 Geometric Parameter Set #1

Figures 23 - 25 show the influence of each parameter in geometric set #1 on vehicle RPER at each of the three speeds. At an initial side velocity of 22 Ft/S Figure 23 shows that the track width (TRW) and the distance from the axle to the static sprung mass c.g. position (h_s) are the most influential elements of geometrical parameter set #1 and exhibit opposite influence on vehicle RPER. A positive value of sensitivity taken with respect to TRW indicates that increasing a vehicle's track width increases its RPER, while a negative value of sensitivity taken with respect to h_s indicates that increasing the height of the sprung mass c.g. position above the axle centerline decreases vehicle RPER. The line labeled SRSF represents the static rollover stability factor which is equal to the half of the track width divided by the center of gravity height. SRSF is a secondary parameter which was substituted into the model where the ratio of track width to c.g. height was found. The sensitivity of RPER to this secondary parameter was found to be a constant positive value which was independent on the initial velocity. These results suggest that a vehicle RPER can be increased by increasing its static rollover stability factor.

Since the initial side velocity of 22 Ft/S did not result in rollover the vehicle rocks back and forth after impact. Because of this periodic motion, the sensitivity of parameters TRW and h_s oscillate as well, changing signs several times during the last half of the sensitivity run.

Overall, Figure 23 shows that a vehicle's RPER can be increased by either increasing its track width, or by reducing the static height of the sprung mass c.g. above the axle center line, which makes sense from a practical viewpoint. The figure shows that parameters h_u, h_r, L, and a do not appear to be influential in this case.

Figures 24 and 25 show the percentage sensitivity functions for vehicle/curb impacts with initial lateral velocities of 23.075 Ft/S and 25 Ft/S, speeds which result in vehicle rollover. These figures illustrate that track width (TRW) and static height of the sprung mass above the axle centerline (h_s) are still the most influential parameters among those shown in the figures, and that vehicle RPER can be increased by increasing TRW or reducing h_s.

Because the initial side velocities of 23.075 and 25 Ft/S result in rollover the vehicle does not rock back and forth after impact as it did using an initial velocity of 22 Ft/S. For this reason the percentage sensitivity functions do not oscillate in the rollover cases.

6.2 Geometric Parameter Set #2

Figures 26 - 28 illustrate the influence of geometric parameter set #2 on vehicle RPER using percentage sensitivity fun-

ctions at the different initial lateral velocities. Each of the figures show that the initial distance from the curb Y_u is the most influential parameter. This is because changing the initial distance from the curb changes the time of impact with the curb as well as speed of impact. This results in the spikes seen in Figures 26 - 28. Since the initial distance from the curb is not a vehicle design parameter but rather a parameter dependent on circumstance its importance is not relevant and can be ignored. With this in mind, Figure 26 (obtained using a speed of 22 Ft/S) indicates that the tire radius T_r is very important. This finding is significant because an increase in tire radius increases the c.g. height of the entire vehicle. The figure shows that increasing the tire radius decreases a vehicle's RPER. The figure also shows that the undeflected spring length SPRLNG is important and that increasing a vehicle's undeflected spring length decreases its RPER. This makes sense because in STI's model an increase in SPRLNG raises the c.g. height of the sprung mass. As before, the oscillation in the sensitivity functions which appear after 1 second are due to the periodic roll motion which occurs in the non-rollover cases. Figure 26 also suggests that vehicle RPER can be raised by increasing the height of the curb.

The influence of elements making up geometric parameter set #2 on vehicle RPER at the rollover speeds of 23.075 and 25 Ft/S are shown in Figures 27 and 28. These figures also reveal that a vehicle's RPER can be increased by increasing the height of the curb, decreasing the tire radius, or decreasing the undeflected

spring length. The parameters s and ABSDEF appeared to be relatively unimportant.

6.3 Mass Parameter Set

Figures 29 - 31 show the influence of parameters associated with vehicle mass properties on vehicle RPER. When these figures are compared with Figures 25 - 28 it is evident that, as a whole, the mass parameters exhibit less influence on the system response than geometric parameters since the magnitudes of the sensitivity functions for the mass parameters are much smaller up until vehicle rollover occurs.

Figure 29 shows the percentage sensitivity functions of vehicle RPER taken with respect to each mass parameter using an initial lateral velocity of 22 Ft/S. The figure shows that vehicle RPER can be raised by increasing vehicle sprung mass and, to a lesser extent, vehicle unsprung mass. The periodic rolling motion common to the non-rollover case causes an oscillation in the sprung mass sensitivity of vehicle RPER at 1.2 seconds. Parameters associated with vehicle inertia exhibit virtually no sensitivity until the point of impact. This is reasonable since the angular velocity of the vehicle is nearly zero up until impact with the curb. After impact the sensitivity values taken with respect to inertial parameters remain smaller than those for vehicle mass.

The sensitivity results obtained at the critical rollover speed of 23.075 Ft/S (Fig. 30) reveal that, initially, vehicle RPER can be increased by increasing sprung and unsprung masses. Later however, the sensitivity of the sprung mass changes sign. This may be due to the fact that at small roll angles the gravitational force acting on the sprung mass tends to return the vehicle to a four wheel stance and is hence a stabilizing force. However at larger roll angles (angles at which the line of action of the gravitational force on the sprung mass acts outside the vehicle's track width) the gravitational force tends to pull the vehicle over and is a destabilizing force. Again the sensitivity values taken with respect to the inertial parameters are zero up until the point of impact. After impact the sensitivity of the inertial parameters remain small until the sensitivity functions diverge at 1.3 seconds at which time they exhibit positive sensitivity, indicating that vehicle RPER can be increased by increasing vehicle inertia properties. This is logical since inertial forces tend to resist vehicle motion. The divergent nature of the curves is due to large scale differences in system response caused by small parameter changes at a speed which is very near the threshold of rollover.

The sensitivity of vehicle RPER taken with respect to vehicle mass parameters using an initial side velocity of 25 Ft/S (Fig. 31) show trends which are similar to those obtained using initial side velocities of 22 and 23.075 Ft/S. Again, vehicle RPER can be increased, at least initially, by increasing sprung and unsprung masses. After impact the sensitivity of the sprung

mass parameter changes sign. The sensitivity of vehicle RPER to changes in inertial parameters remain zero up until the time of impact, after which time the inertial parameters exhibit small positive sensitivity.

6.4 Stiffness Parameter Set

Figures 32 - 34 illustrate the sensitivity of vehicle RPER to changes in vehicle stiffness parameters. A comparison of these figures with those for geometrical and mass properties (Fig. 23 - 31) reveals that the variation of vehicle stiffness parameters has less of an overall effect on vehicle RPER than variation of geometrical and mass parameters.

Figure 32 shows the influence of stiffness parameters on vehicle RPER using an initial side velocity of 22 Ft/S. The figure shows that the variation of stiffness parameters have little effect on vehicle RPER. The only two parameters which exhibit some significant influence are suspension spring stiffness K_1, which is negative, and the wheel compliance KD1 which is positive.

At the critical rollover velocity of 23.075 Ft/S the only stiffness parameter which exhibits a significant influence on vehicle RPER is suspension spring stiffness K1 (Fig. 33). The divergent nature of the sensitivity functions at a time equal 1.3

seconds is again due to large scale differences in system response caused by small parameter changes at a speed which is very close to the rollover threshold.

At the initial lateral velocity of 25 Ft/S the percentage sensitivity of vehicle RPER to stiffness parameter KD2 reaches a relatively large positive value just after impact, suggesting that a vehicle's rollover propensity could be effected by the vehicle's force/deformation relationship.

6.5 Damping Parameter Set

Figures 35 - 37 illustrate the effect of damping parameter variation on vehicle RPER. These figures indicate that, as a whole, damping parameters have much less influence on RPER than geometrical, mass, and stiffness parameters. The shock absorber damping coefficient B_1 had the greatest amount of influence among all of the damping parameters examined, however its sensitivity value was small and oscillated between positive and negative values making it difficult to draw any conclusions regarding its effect in tripped rollover situations.

6.6 Force Parameter Set

The influence of force related parameters on vehicle RPER is shown in Figures 39 - 40. The sliding coefficient of friction

between the tire and pavement is the most influential parameter of the set. This was expected since this coefficient affects the time at which the vehicle strikes the curb as well as the speed of impact. A higher coefficient of friction delays the time of contact and reduces the impact velocity, resulting in a large positive value of the sensitivity function at the instant of contact.

At each of the three speeds the sensitivity of deformation parameter POINT1 and frictional saturation speed K_V are quite small. The deformation parameter POINT2 does not exhibit any influence in the case of the initial side velocity of 22 Ft/S, suggesting that the impact deformation never reaches the third deformation region. However, in the 23.075 and 25 Ft/S cases POINT2 exhibits strong positive sensitivity immediately after impact, suggesting that vehicle RPER can be increased if the plastic impact deformation is spread over a large distance. By spreading the deformation over a large distance the amount of work done crushing the vehicle is increased, reducing the amount of energy which can contribute to the vehicle's rolling motion.

The constant force of plastic deformation WHLDEF exhibited some influence in all three cases, however it was difficult to draw any firm conclusions regarding its effect on vehicle rollover propensity.

6.7 Detailed Investigation of Influential Parameters

Once all of the vehicle's primary parameters had been analyzed, those which exhibited considerable influence on vehicle response were analyzed in detail. The percentage sensitivity functions of vehicle RPER were computed using 5 different initial side velocities (17, 22, 23.075, 25, and 27 Ft/S). As before the initial distance from the front inside wheel to the curb was equal to 7.5 feet and the vehicle's initial heading angle was equal to zero. The sensitivity functions were computed for the first 0.8 seconds of vehicle motion in order to determine what takes place during the initial phases of the tripping maneuver. The six parameters which were investigated are, vehicle track width (TRW), sprung mass (M_s), tire radius (T_r), the horizontal distance from the front axle to the vehicle c.g. location (a), the static distance from the sprung mass c.g. position to the axle center line (h_s), and the distance from the axle center line to the lateral restraining force (h_r).

Figure 41 shows the sensitivity of RPER taken with respect to the six parameters using an initial side velocity of 17 Ft/S. At this speed the vehicle comes to a complete stop prior to impact with the curb. The sensitivity results show that track width is the most influential parameter, followed by sprung mass M_s and distance h_s. Parameters T_r, h_r, and a exhibited very little influence on RPER.

The sensitivity obtained using a initial side velocity of 22 Ft/S is shown on Figure 42. The sensitivity functions in this figure are identical to those obtained for a speed of 17 Ft/S up until the impact time of 0.5 seconds. After impact track width becomes more important as does h_s and T_r. Parameters a and h_r exhibit little influence on the vehicle RPER. The sensitivity results generated using an initial velocity of 23.075 (Fig. 43) continue this trend.

At 25 Ft/S vehicle track width is still the most influential parameter (Fig. 44), followed by tire radius T_r and the distance from the static sprung mass c.g. position to the axle center line (h_s). Sprung mass also becomes important, changing its sensitivity function from positive to negative right after impact. Parameters a and h_r still remain relatively unimportant. These trends are amplified in the case using an initial side velocity of 27 Ft/S (Fig. 45).

6.8 System Response Sensitivity

The influence of parameters TRW, a, L, M_s, I_z, K_1, and B_1 on vehicle system response (using an initial side velocity of 23.075 Ft/S) are shown in Figures 46 - 53. The influence of these parameters on the sprung mass roll angle is shown in Figure 46. After curb impact (which occurs at approximately 0.45 seconds) the figure illustrates that sprung mass roll angle can be decreased by increasing vehicle track width (TRW), roll inertia of

the sprung mass (I_s), and the horizontal distance from the front axle to vehicle c.g. position (a), while increases in vehicle sprung mass (M_s), tire radius (T_r) and wheelbase (L) will increase the sprung mass roll angle. The suspension damping coefficient (B_1), suspension spring stiffness (K_1), and yaw moment of inertia (I_z) each appear to have little influence on the sprung mass roll angle.

Figures 47 and 48 show the influence of the same parameter set on sprung mass roll velocity and sprung mass roll acceleration. The interpretation of these figures agree with that of Figure 46, however it is made more difficult by the discontinuous nature of the sensitivity curves.

The influence of this parameter set on the roll angle and roll angular velocity of the unsprung mass are shown in Figures 49 and 50. The conclusions which can be draw from these figures are quite similar to those obtained from Figure 46. The unsprung mass roll angle and angular velocity can be decreased by increasing vehicle track width (TRW), sprung mass roll moment of inertia (I_s), and the horizontal distance from the front axle to the vehicle c.g. position (a), while increases in vehicle sprung mass (M_s), tire radius (T_r) and wheelbase (L) result in increases in the unsprung mass roll angle and angular velocity.

The percentage sensitivity functions of the sprung and unsprung mass c.g. heights to parameter changes are shown in fig-

ures 51 & 52. It should be noted that the reference system used by STI's model has its positive Z direction vector pointing into the ground. Therefore a positive value of percentage sensitivity indicates that increasing the parameter value will decrease the actual distance from the center of gravity position to the ground. With this in mind the figures show that the distance from the sprung mass c.g. positions to the ground can be increased by increasing tire radius (T_r), spring stiffness (K_1), and after impact track width (TRW), while the distance from the unsprung mass c.g. position to the ground can be increased primarily by increasing tire radius (T_r). Increasing the sprung mass (M_s) will decrease the distance from the sprung mass c.g. position to the ground. It appears that the parameters a, L, I_s, I_z, and B_1 have little influence on the c.g. heights of both sprung and unsprung masses.

The influence of the parameters on sprung mass vertical velocity is shown in Figure 53. The sensitivity functions taken with respect to parameters T_r and M_s oscillate between positive and negative values which makes the interpretation of results difficult. Parameters a, L, I_s, I_z, K_1, and B_1 all have little influence on the vertical velocity of the sprung mass.

6.9 Sensitivity of RPER to System Response

The final sensitivity functions presented in this report investigate the influence of system state variables on vehicle

RPER. Figures 54 - 58 show how the variation of Z_s, Z_u, Z_s, Z_u, Y_s, and Y_u affects RPER at the initial lateral velocities of 17, 22, 23.075, 25, and 27 Ft/S. At each speed the figures indicate that the state variables Z_s, Z_u, Y_s, and Y_u have little influence on vehicle RPER, however, the figures do show that vehicle RPER can be increased by reducing the sprung and unsprung center of gravity heights Z_s and Z_u.

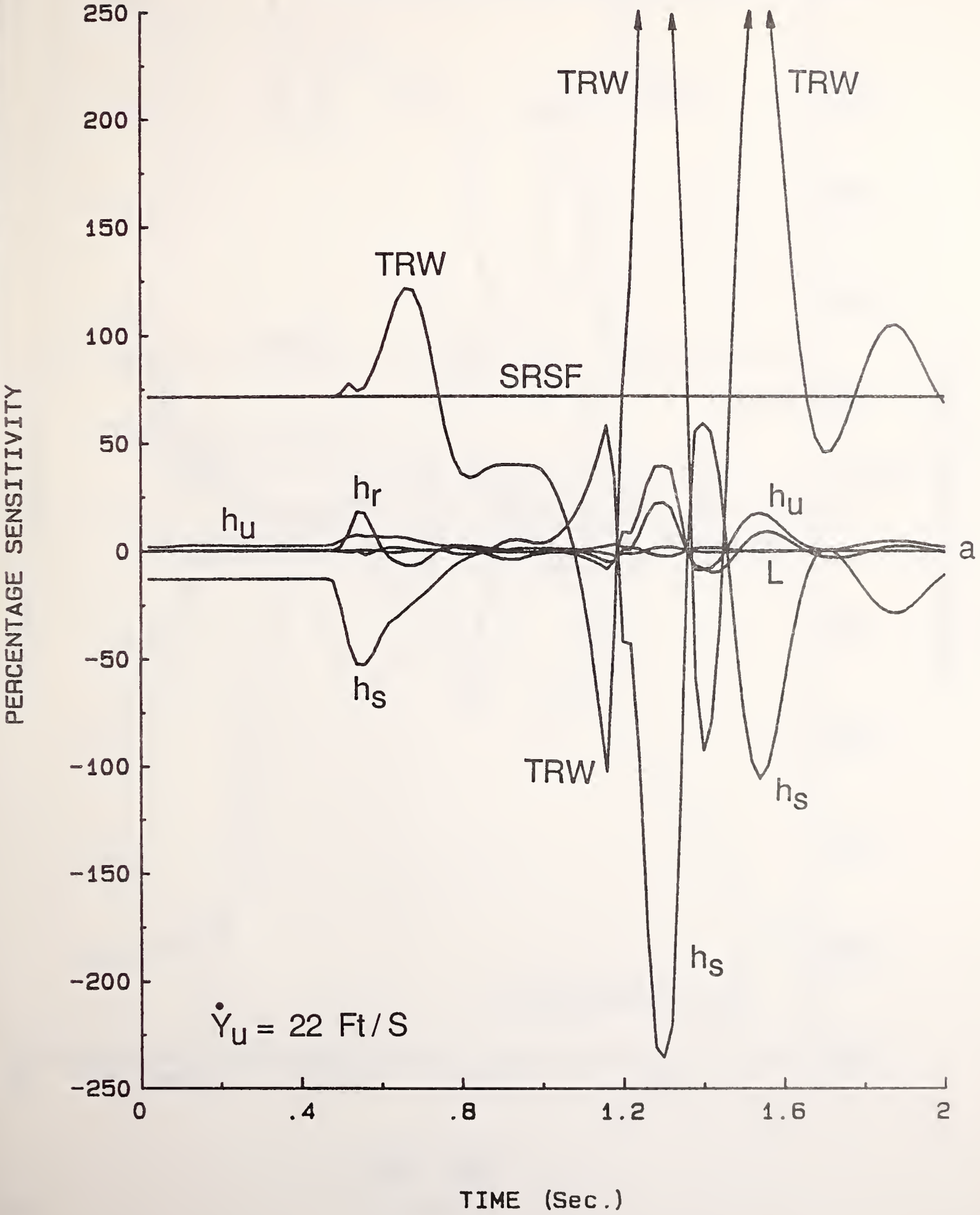

SENSITIVITY OF RP ENERGY RESERVE
GEOMETRIC PARAMETER SET #1
PERCENTAGE SENSITIVITY
TIME (Sec.)
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
TRW
TRW
TRW
SRSF
h_u
h_r
h_u
a
L
h_s
TRW
h_s
h_s
$\dot{Y}_u$ = 22 Ft/S

Fig. 23

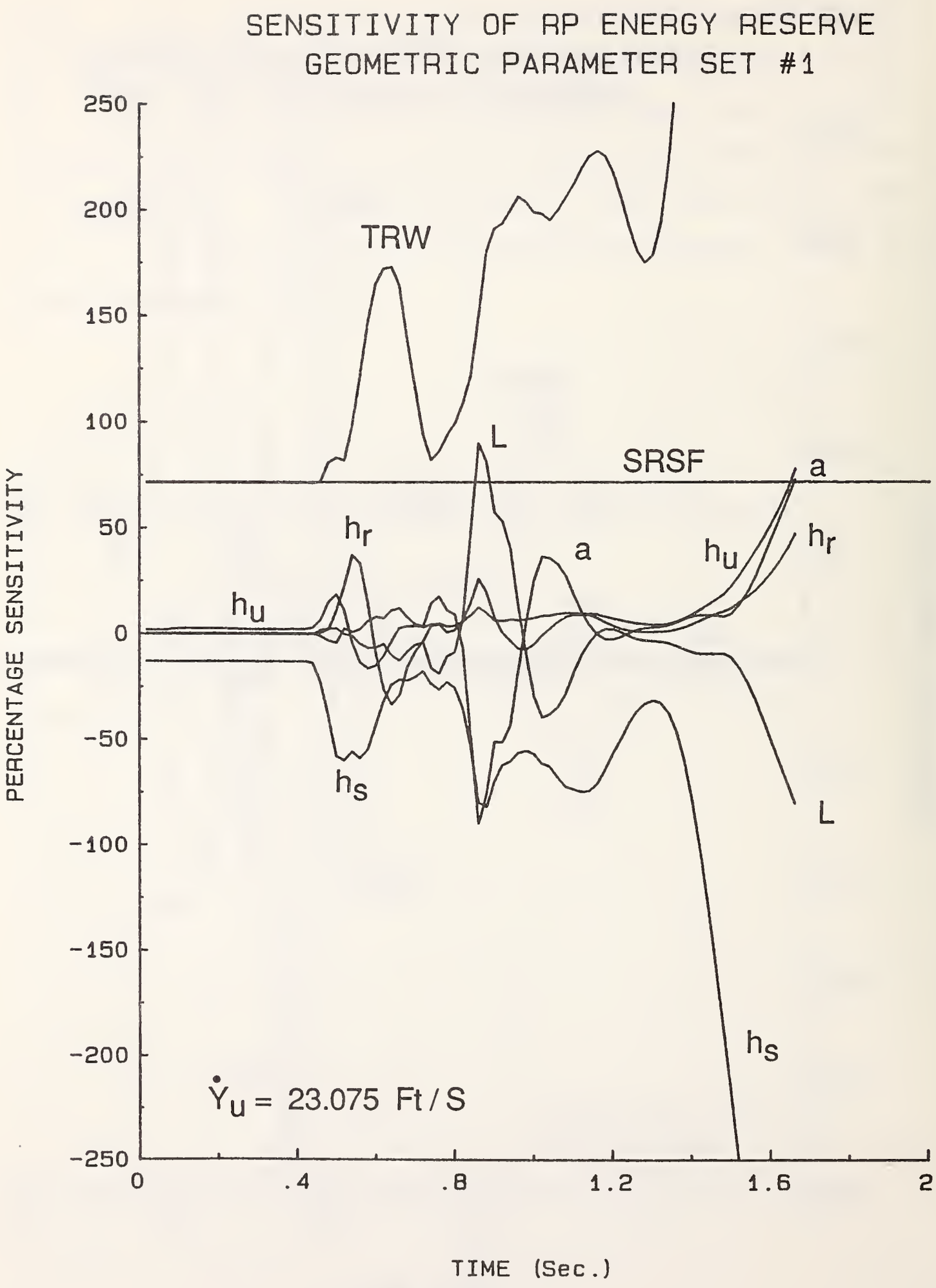
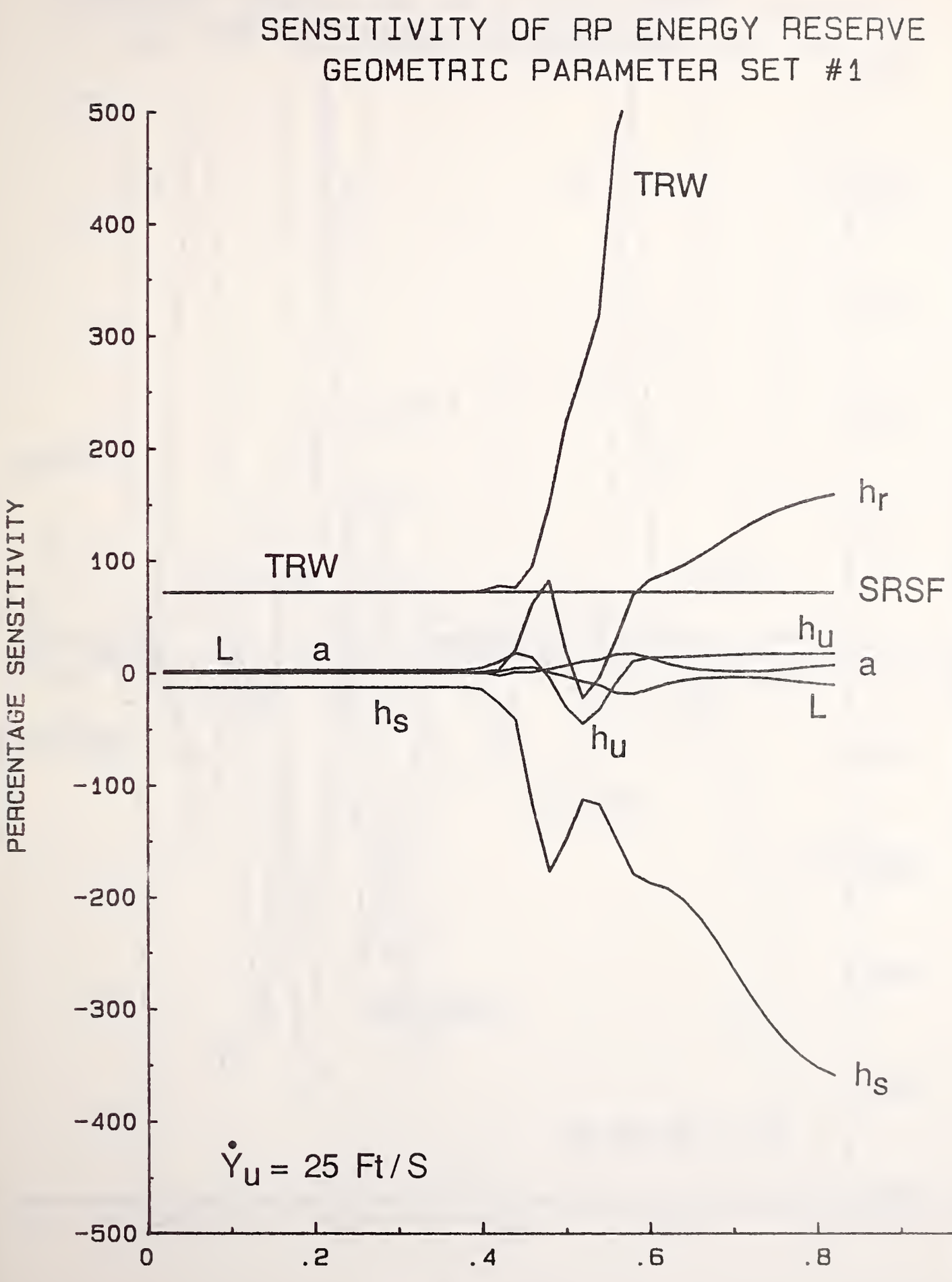

SENSITIVITY OF RP ENERGY RESERVE
GEOMETRIC PARAMETER SET #1
PERCENTAGE SENSITIVITY
TIME (Sec.)
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
TRW
L
SRSF
a
h_r
h_u
h_s
$\dot{Y}_u$ = 23.075 Ft/S

Fig. 24


SENSITIVITY OF RP ENERGY RESERVE
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PERCENTAGE SENSITIVITY
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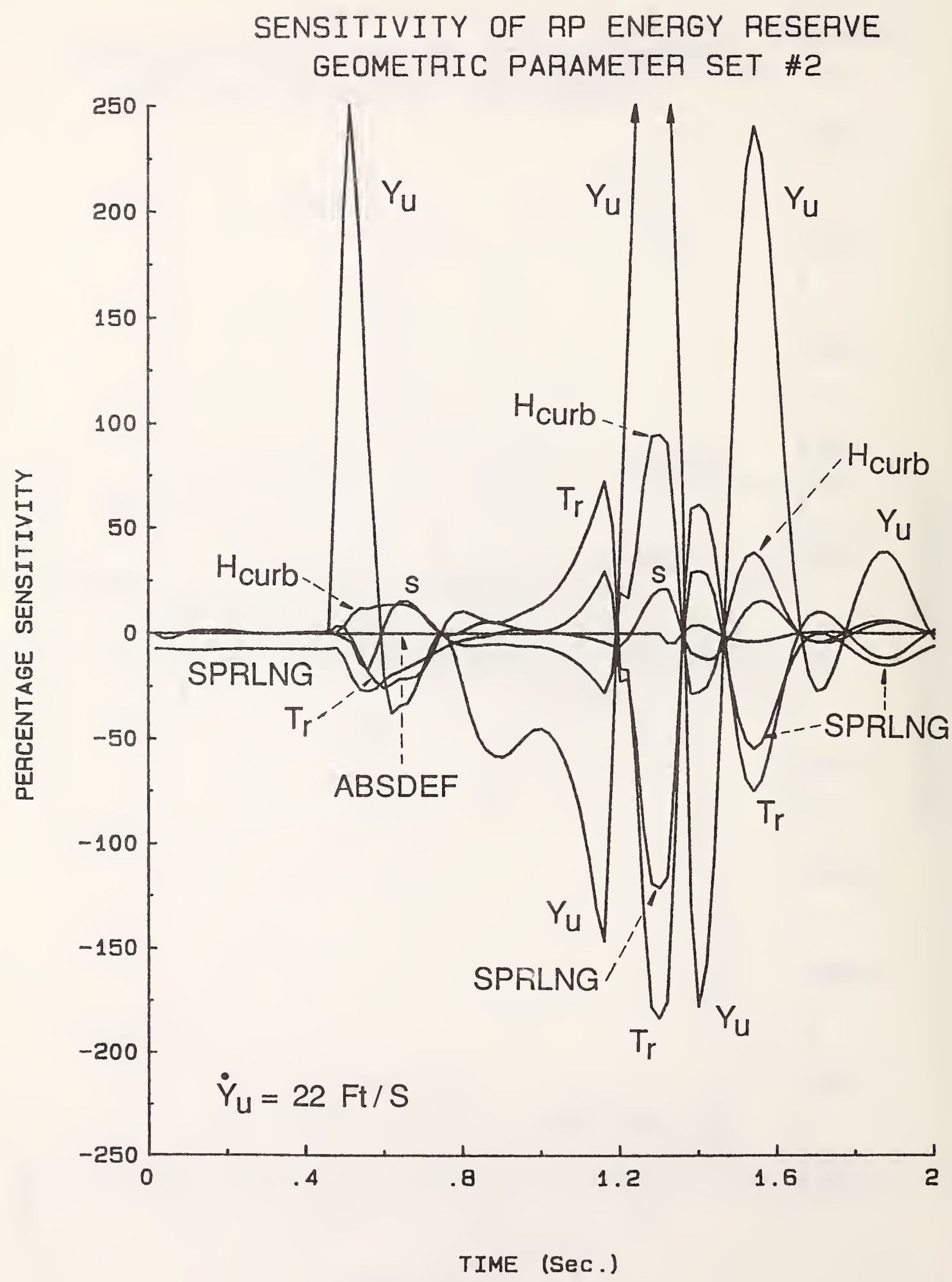
300
200
100
0
-100
-200
-300
-400
-500
0
.2
.4
.6
.8
1
TIME (Sec.)
TRW
TRW
SRSF
h_r
h_u
a
L
L
a
h_s
h_u
h_s
$\dot{Y}_u$ = 25 Ft/S

Fig. 25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GEOMETRIC PARAMETER SET #2
PERCENTAGE SENSITIVITY
TIME (Sec.)
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200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
Yu
Yu
Yu
Hcurb
Hcurb
Hcurb
Tr
Yu
s
s
SPRLNG
Tr
ABSDEF
SPRLNG
Tr
Yu
SPRLNG
Tr
Yu
Ẏu = 22 Ft/S

Fig. 26

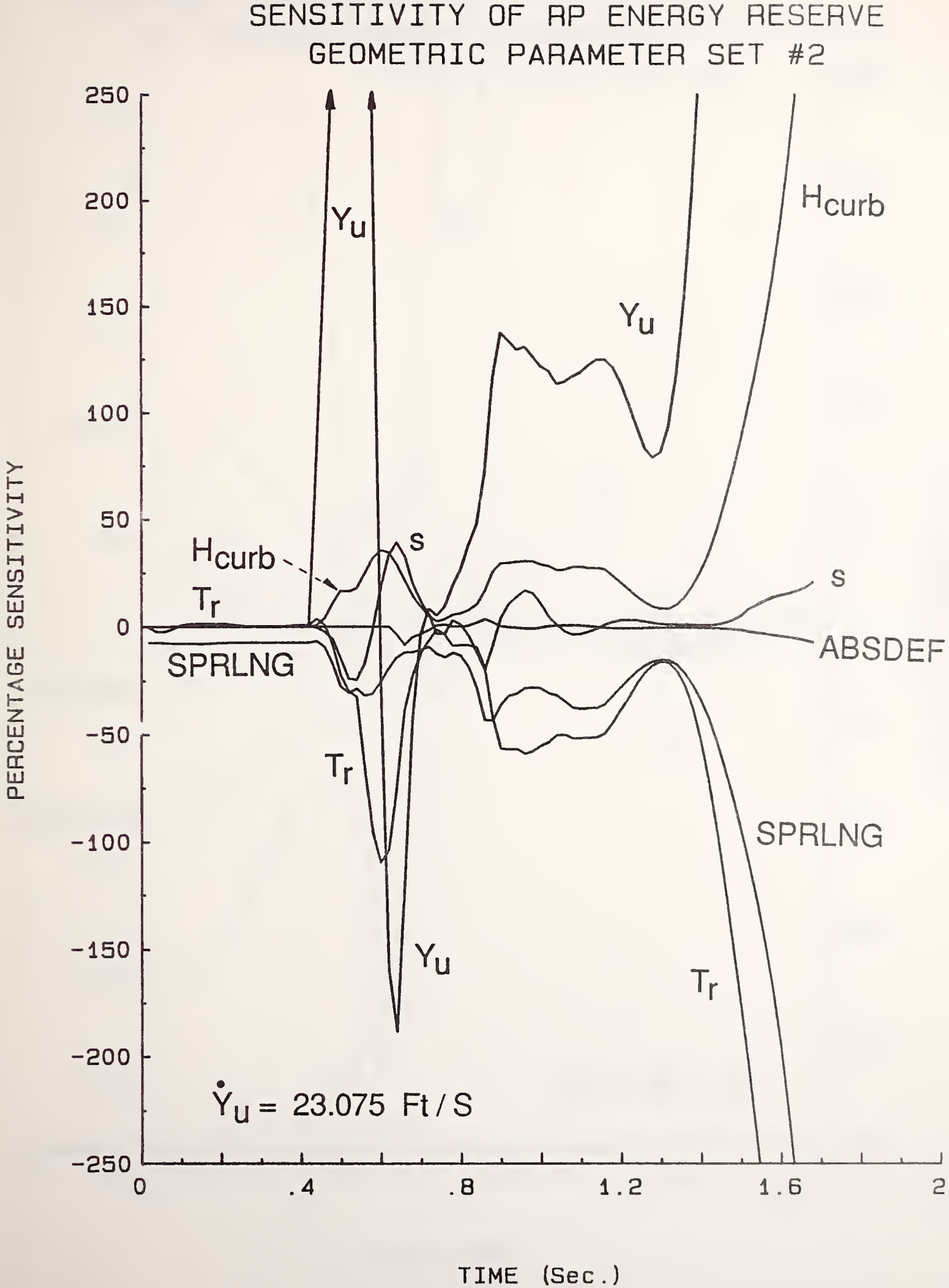
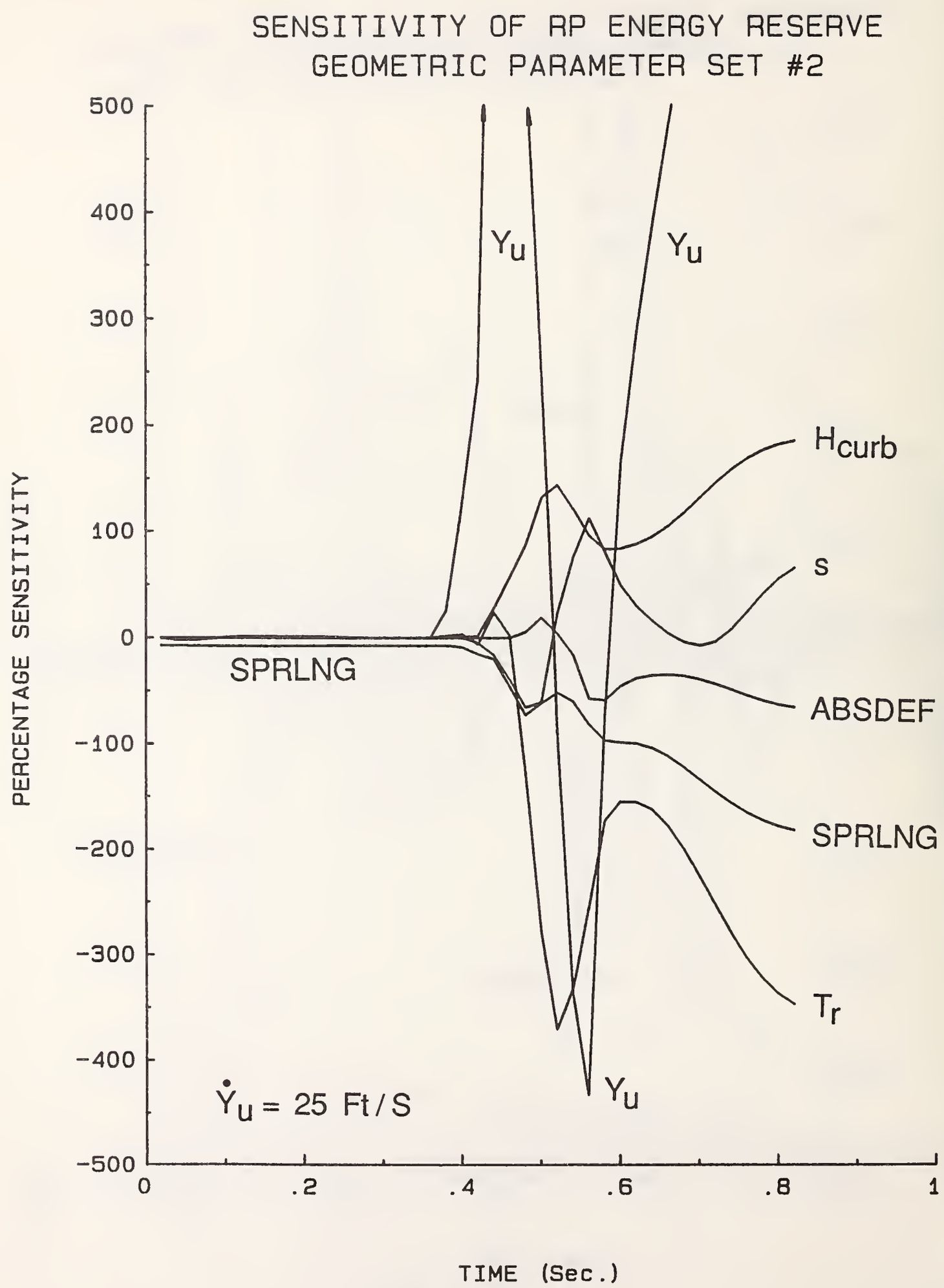

SENSITIVITY OF RP ENERGY RESERVE
GEOMETRIC PARAMETER SET #2
PERCENTAGE SENSITIVITY
TIME (Sec.)
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
Yu
Hcurb
Yu
s
Hcurb
Tr
s
ABSDEF
SPRLNG
Tr
SPRLNG
Yu
Tr
Ẏu = 23.075 Ft / S

Fig. 27


SENSITIVITY OF RP ENERGY RESERVE
GEOMETRIC PARAMETER SET #2
PERCENTAGE SENSITIVITY
TIME (Sec.)
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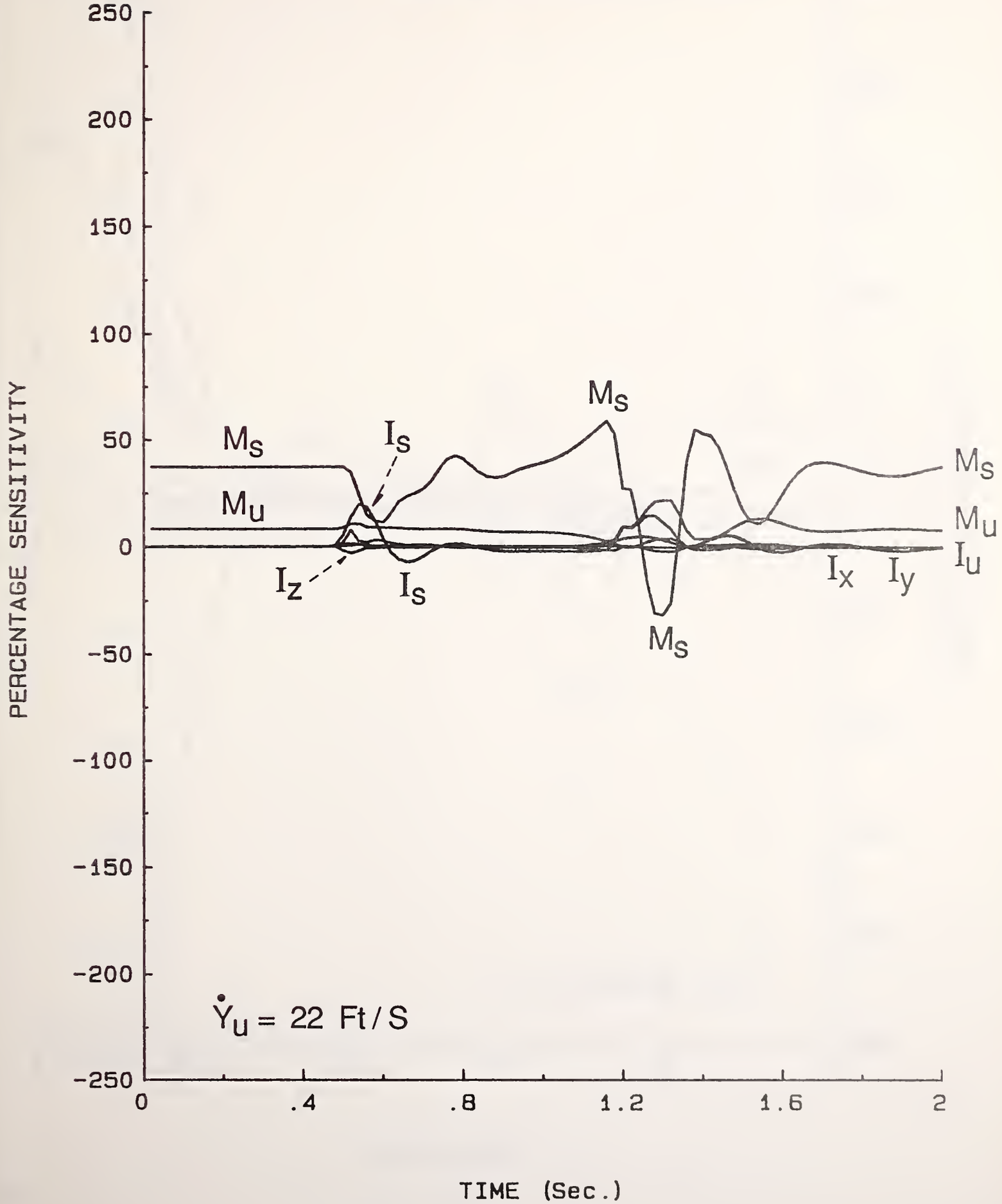
400
300
200
100
0
-100
-200
-300
-400
-500
0
.2
.4
.6
.8
1
Yu
Yu
Yu
Hcurb
s
SPRLNG
ABSDEF
SPRLNG
Tr
Ẏu = 25 Ft/S

Fig. 28


SENSITIVITY OF RP ENERGY RESERVE
MASS PARAMETER SET
PERCENTAGE SENSITIVITY
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200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
TIME (Sec.)
M_S
I_S
M_S
M_S
M_u
I_z
I_S
M_S
I_x
I_y
M_u
I_u
$\dot{Y}_u = 22$ Ft/S

Fig. 29

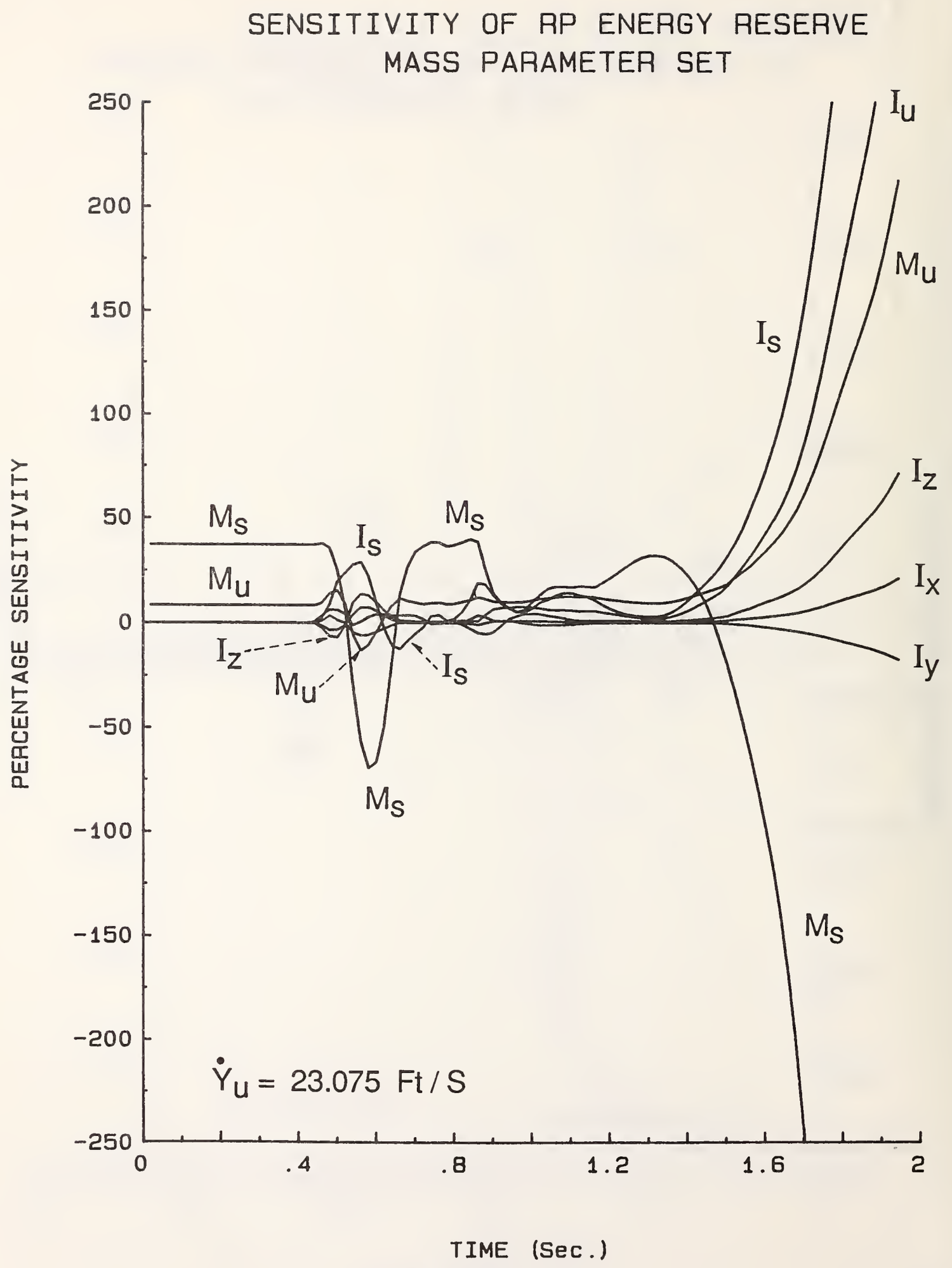
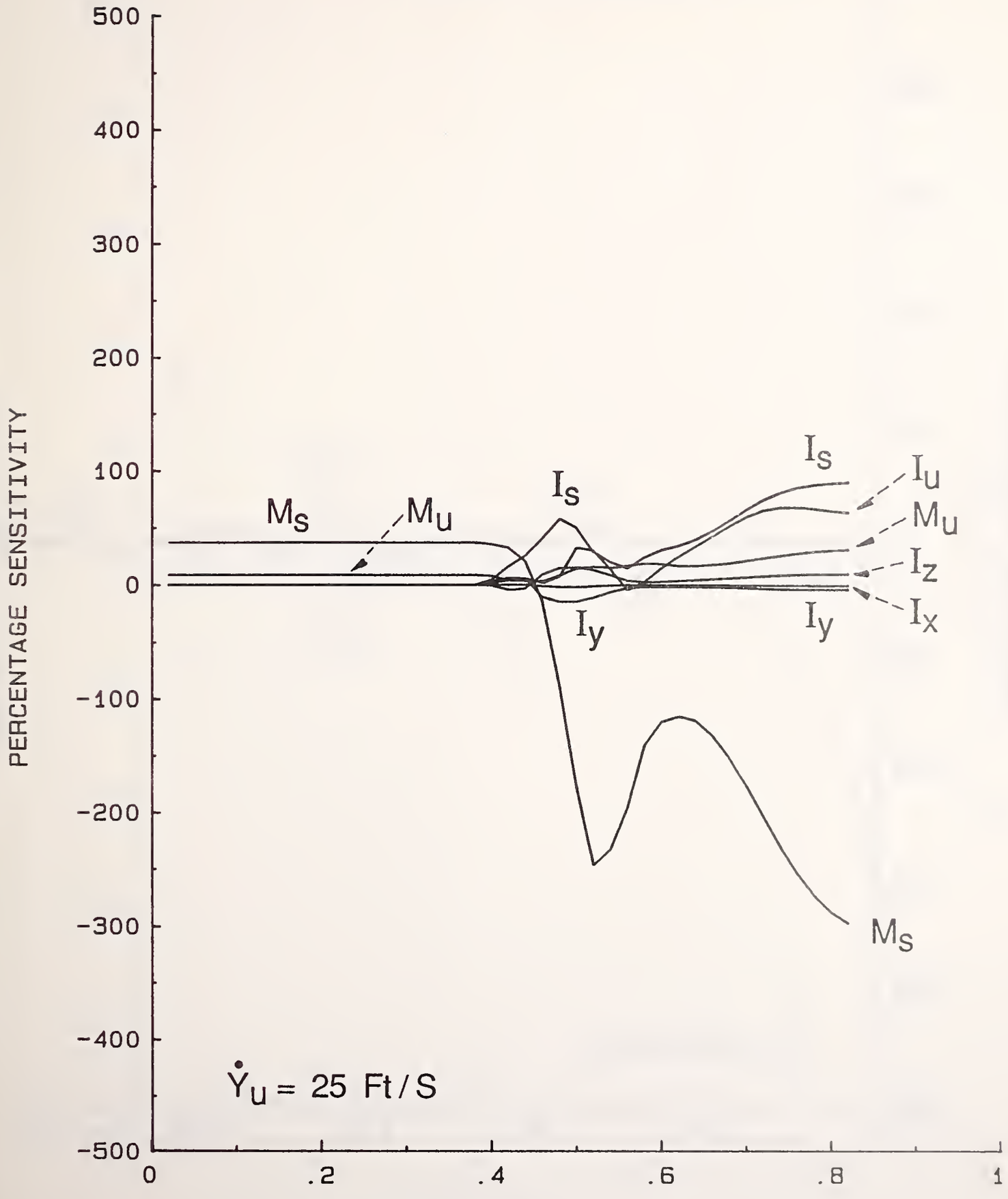

SENSITIVITY OF RP ENERGY RESERVE
MASS PARAMETER SET
PERCENTAGE SENSITIVITY
TIME (Sec.)
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
I_u
M_u
I_s
I_z
I_x
I_y
M_s
M_s
I_s
M_u
I_z
M_u
I_s
M_s
M_s
$\dot{Y}_u$ = 23.075 Ft / S

Fig. 30


SENSITIVITY OF RP ENERGY RESERVE
MASS PARAMETER SET
PERCENTAGE SENSITIVITY
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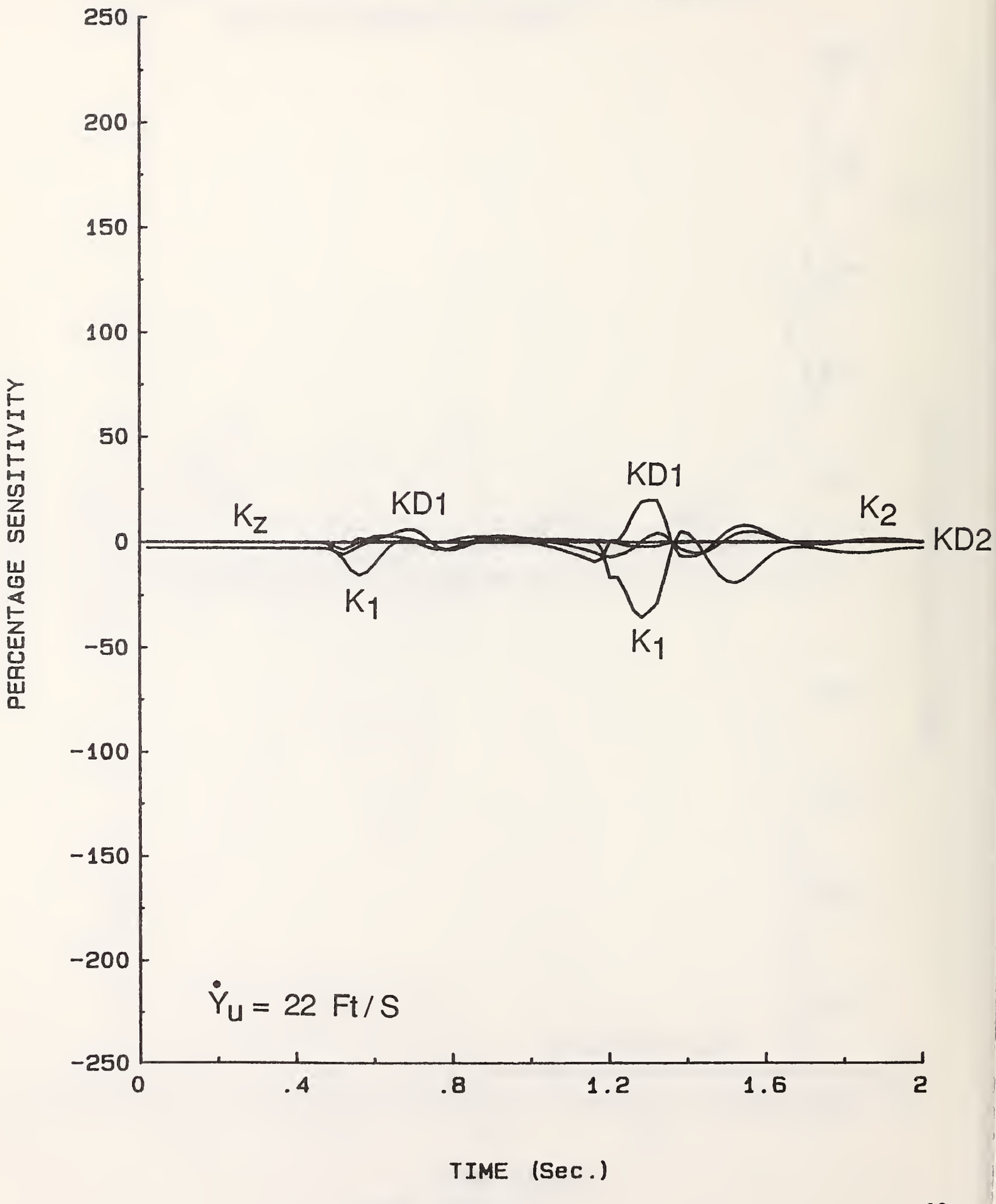
400
300
200
100
0
-100
-200
-300
-400
-500
0
.2
.4
.6
.8
1
TIME (Sec.)
M_S
M_u
I_s
I_y
I_s
I_u
M_u
I_z
I_y
I_x
M_S
$\dot{Y}_u = 25$ Ft/S

Fig. 31


SENSITIVITY OF RP ENERGY RESERVE
STIFFNESS PARAMETER SET
PERCENTAGE SENSITIVITY
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200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
TIME (Sec.)
K_z
KD1
K_1
KD1
K_1
K_2
KD2
$\dot{Y}_u$ = 22 Ft/S

Fig. 32

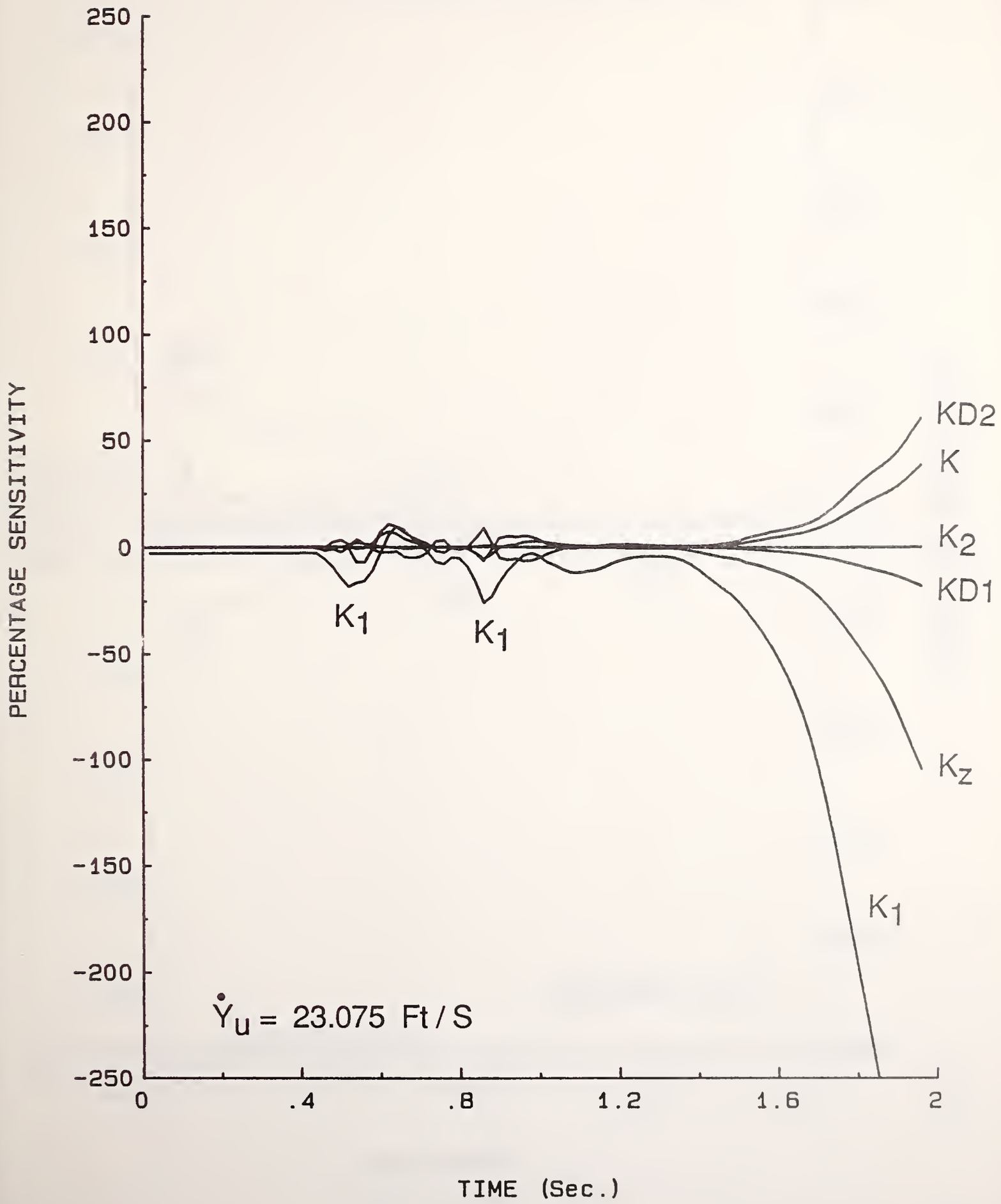
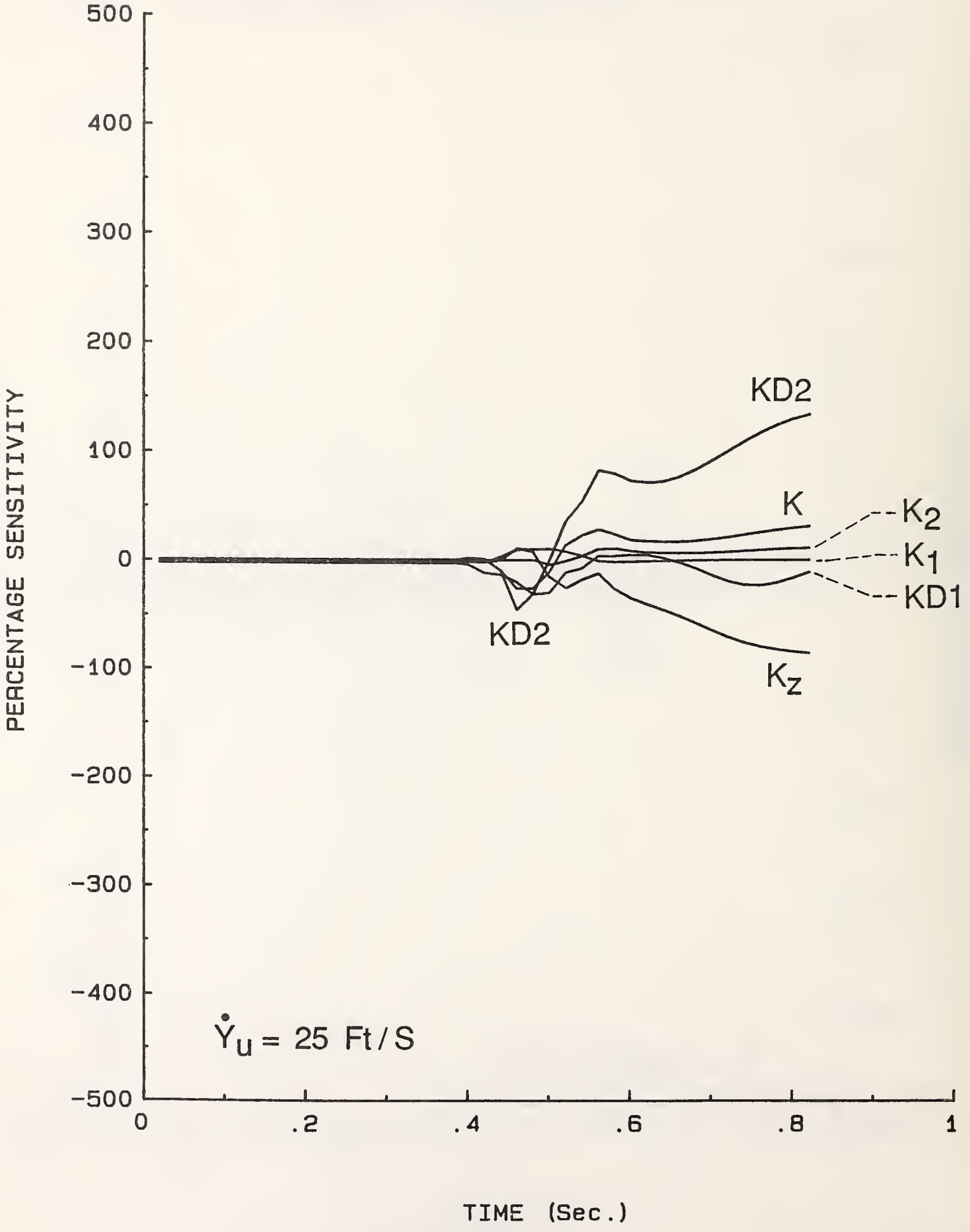

SENSITIVITY OF RP ENERGY RESERVE
STIFFNESS PARAMETER SET
PERCENTAGE SENSITIVITY
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
TIME (Sec.)
KD2
K
K2
KD1
Kz
K1
K1
K1
$\dot{Y}_u$ = 23.075 Ft/S

Fig. 33


SENSITIVITY OF RP ENERGY RESERVE
STIFFNESS PARAMETER SET
PERCENTAGE SENSITIVITY
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400
300
200
100
0
-100
-200
-300
-400
-500
0
.2
.4
.6
.8
1
KD2
K
K2
K1
KD1
KD2
Kz
Ẏu = 25 Ft/S
TIME (Sec.)

Fig. 34

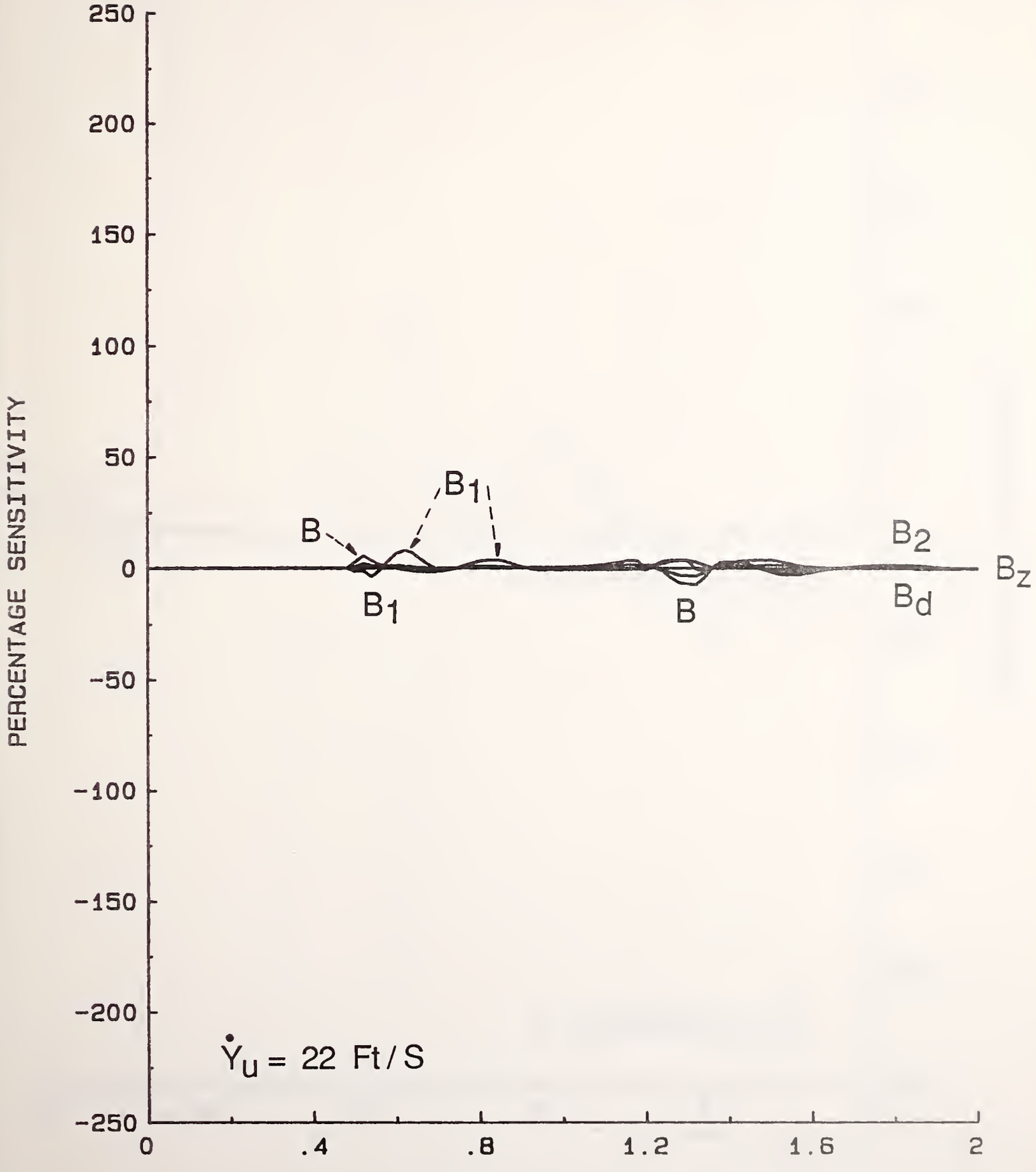
SENSITIVITY OF RP ENERGY RESERVE
DAMPING PARAMETER SET
250
200
150
100
50
0
-50
-100
-150
-200
-250
PERCENTAGE SENSITIVITY
B
B1
B1
B
B2
Bd
Bz
Ẏu = 22 Ft/S
0
.4
.8
1.2
1.6
2
TIME (Sec.)

Fig. 35

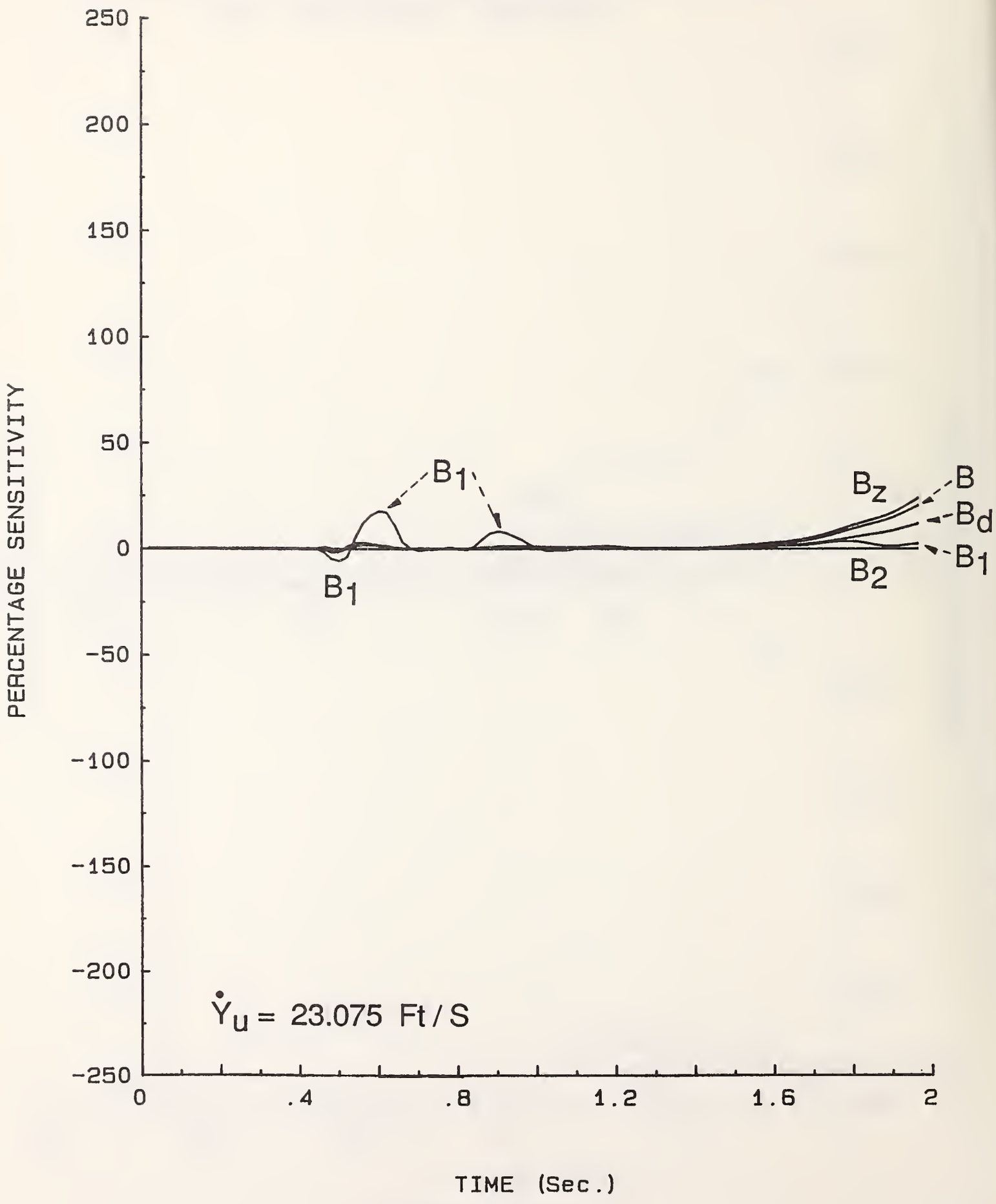
SENSITIVITY OF RP ENERGY RESERVE
DAMPING PARAMETER SET
PERCENTAGE SENSITIVITY
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
TIME (Sec.)
B_1
B_1
B_z
B
B_d
B_1
B_2
$\dot{Y}_u$ = 23.075 Ft / S

Fig. 36

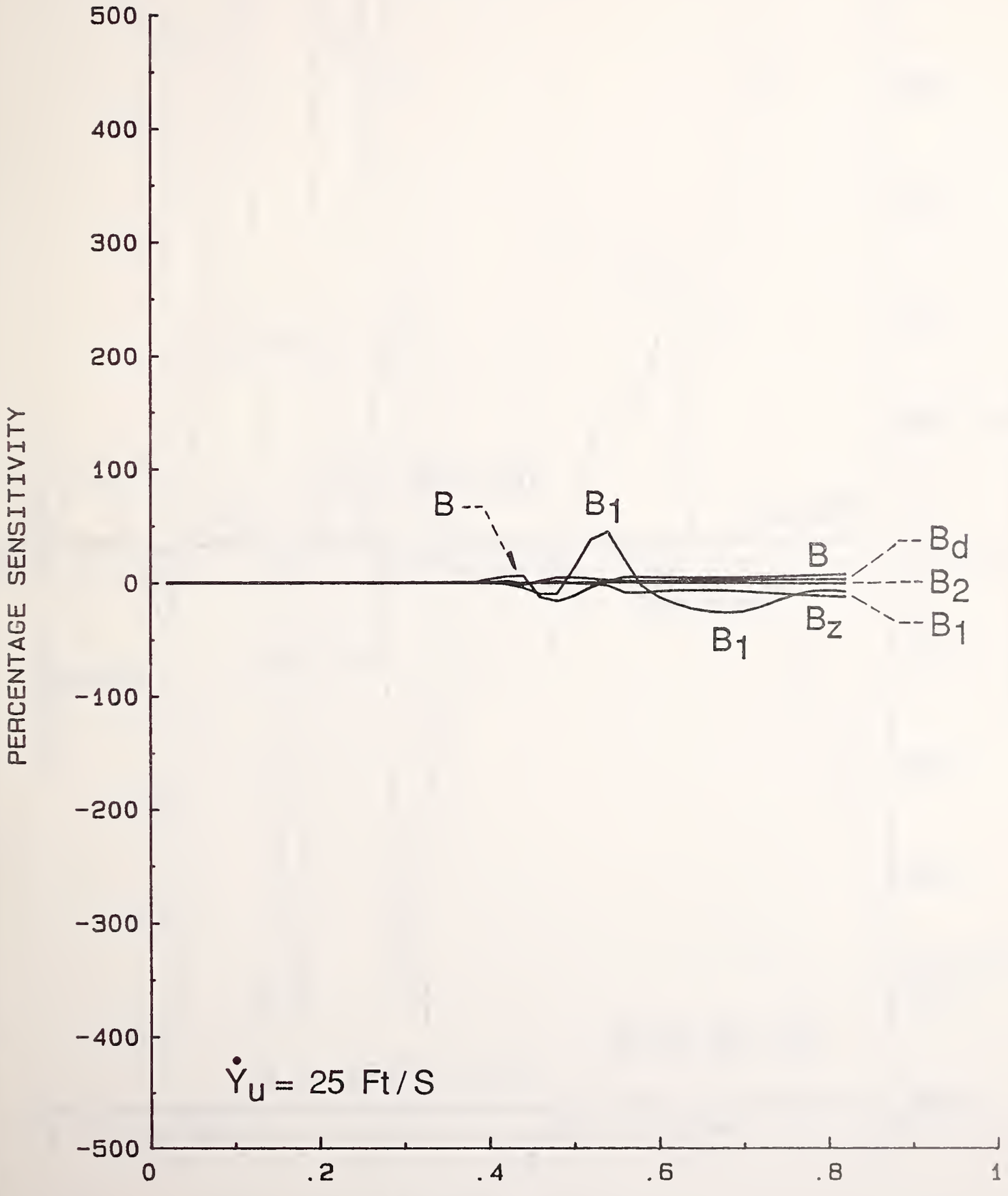
SENSITIVITY OF RP ENERGY RESERVE
DAMPING PARAMETER SET
PERCENTAGE SENSITIVITY
500
400
300
200
100
0
-100
-200
-300
-400
-500
0
.2
.4
.6
.8
1
TIME (Sec.)
B
B1
B
Bd
B2
Bz
B1
B1
Ẏu = 25 Ft/S

Fig. 37

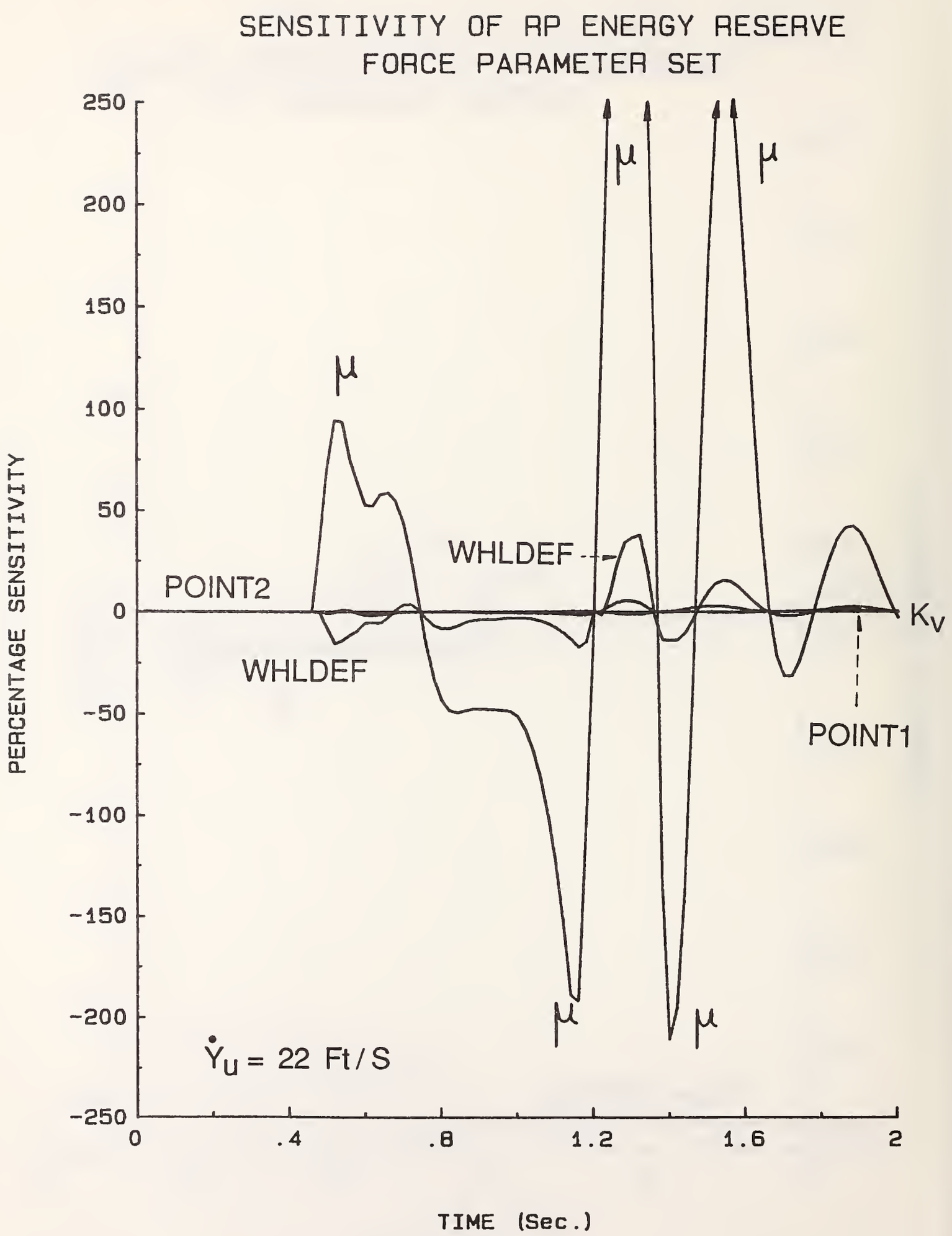
SENSITIVITY OF RP ENERGY RESERVE
FORCE PARAMETER SET
PERCENTAGE SENSITIVITY
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
TIME (Sec.)
μ
μ
μ
μ
μ
μ
WHLDEF
WHLDEF
POINT2
POINT1
K_V
$\dot{Y}_u$ = 22 Ft/S

Fig. 38

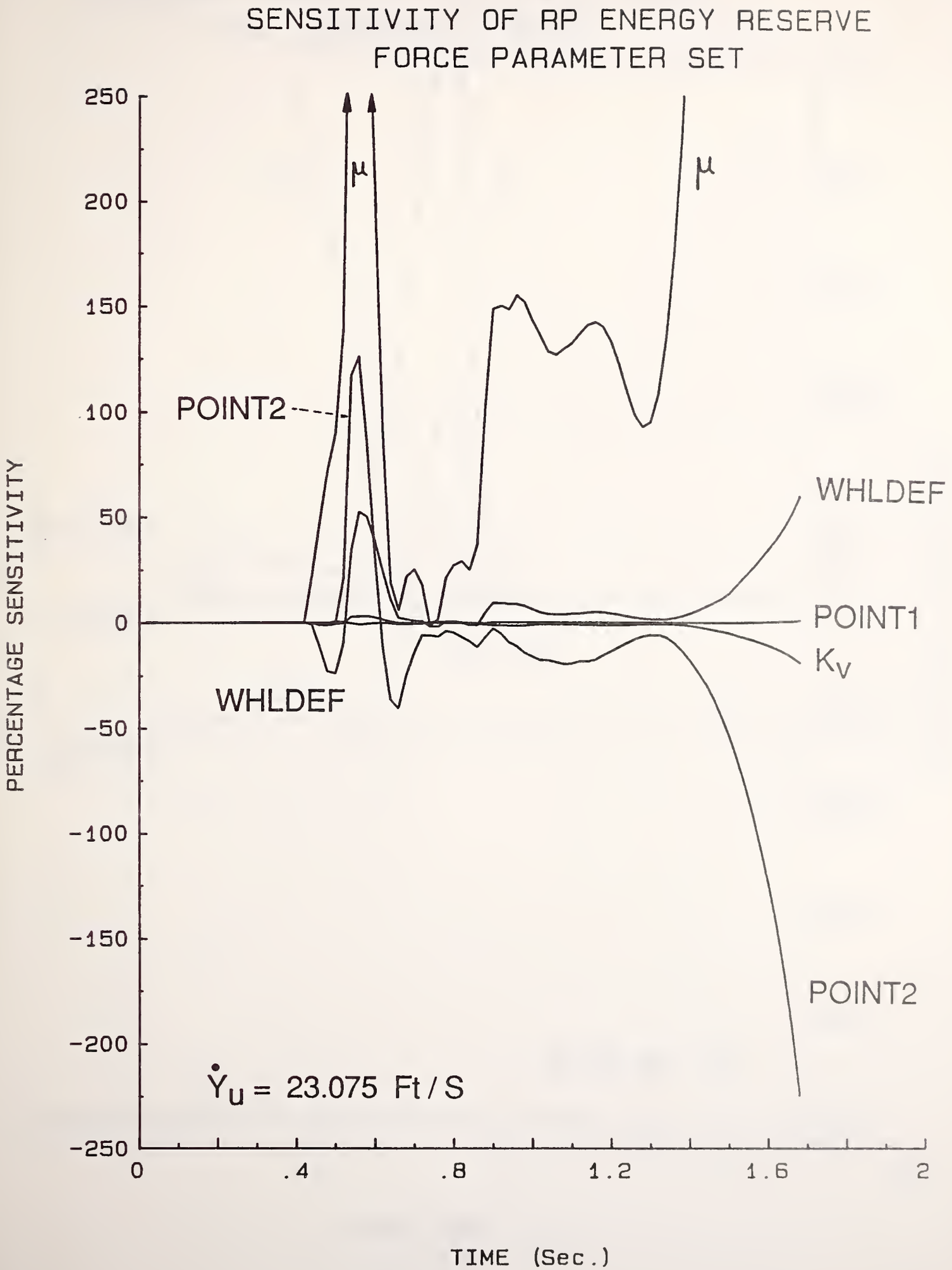
SENSITIVITY OF RP ENERGY RESERVE
FORCE PARAMETER SET
PERCENTAGE SENSITIVITY
250
200
150
100
50
0
-50
-100
-150
-200
-250
0
.4
.8
1.2
1.6
2
TIME (Sec.)
μ
μ
POINT2
WHLDEF
POINT1
K_V
WHLDEF
POINT2
$\dot{Y}_u$ = 23.075 Ft/S

Fig. 39

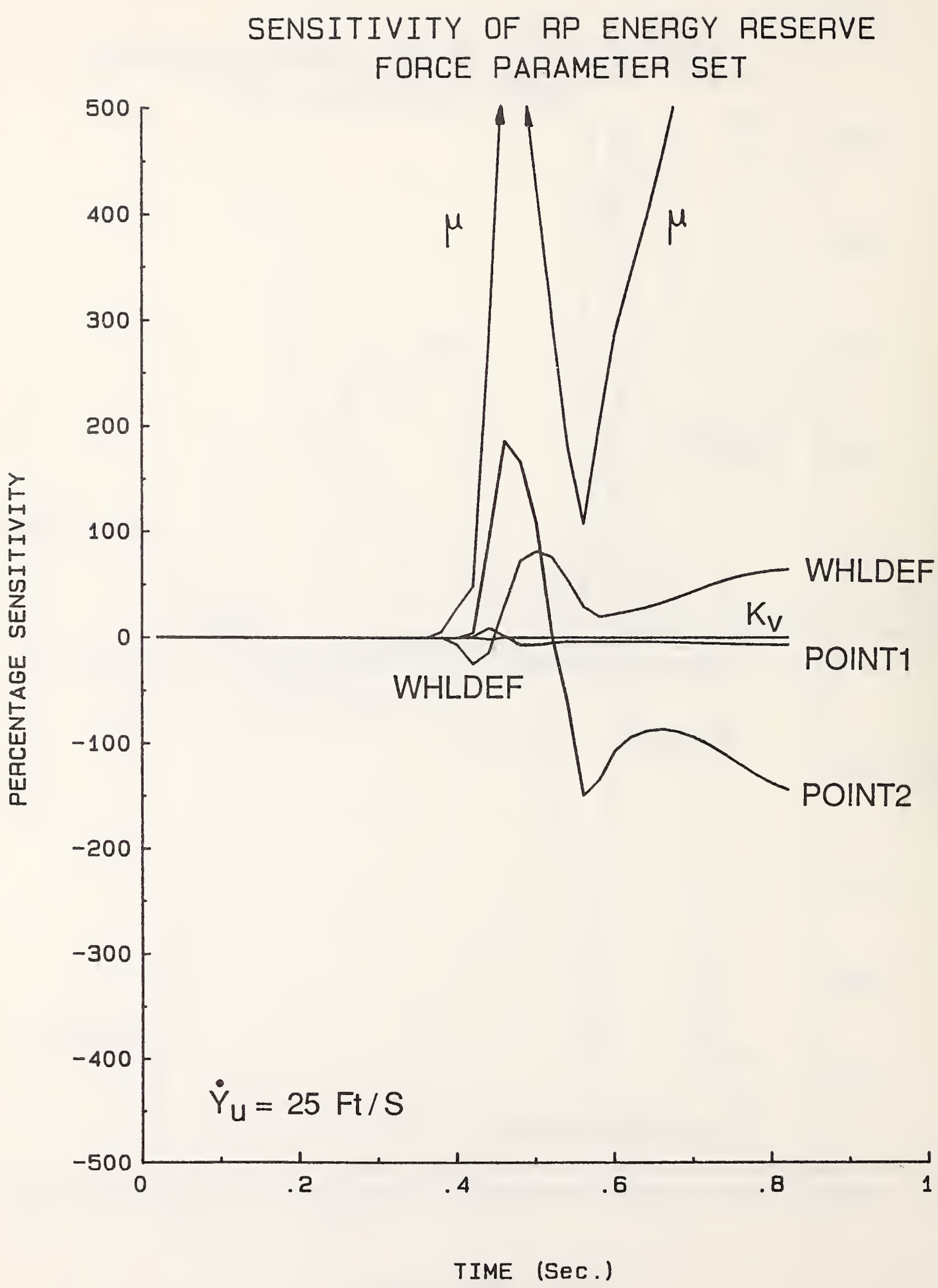
SENSITIVITY OF RP ENERGY RESERVE
FORCE PARAMETER SET
PERCENTAGE SENSITIVITY
500
400
300
200
100
0
-100
-200
-300
-400
-500
0
.2
.4
.6
.8
1
TIME (Sec.)
μ
μ
WHLDEF
K_V
POINT1
WHLDEF
POINT2
$\dot{Y}_u$ = 25 Ft/S

Fig. 40

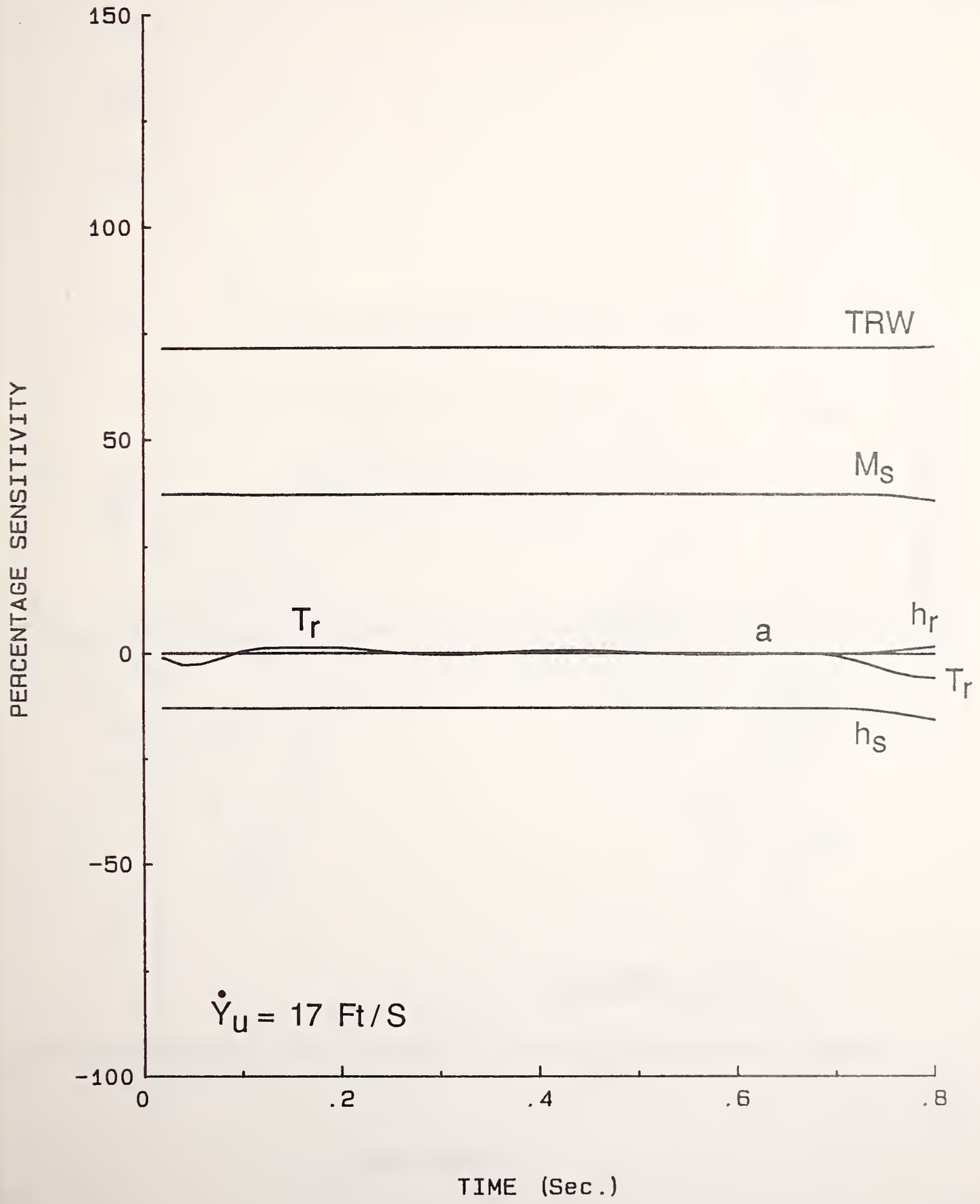
SENSITIVITY OF ROLLOVER
PREVENTION ENERGY RESERVE
150
100
50
0
-50
-100
PERCENTAGE SENSITIVITY
TRW
Ms
Tr
a
hr
Tr
hs
Ẏu = 17 Ft/S
0
.2
.4
.6
.8
TIME (Sec.)

Fig. 41

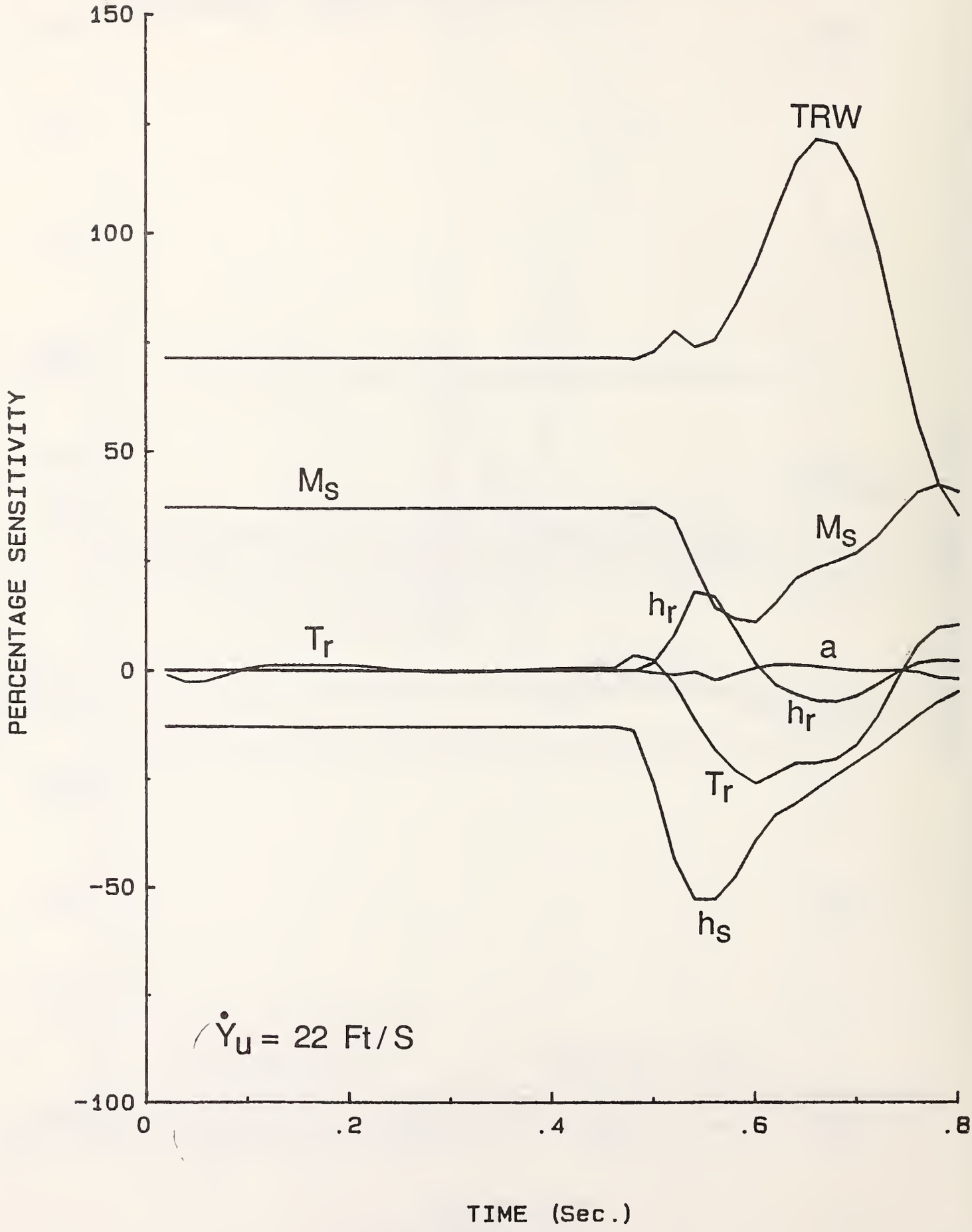
SENSITIVITY OF ROLLOVER
PREVENTION ENERGY RESERVE
150
100
50
0
-50
-100
PERCENTAGE SENSITIVITY
TRW
M_S
M_S
h_r
T_r
a
h_r
T_r
h_S
$\dot{Y}_u$ = 22 Ft/S
0
.2
.4
.6
.8
TIME (Sec.)

Fig. 42

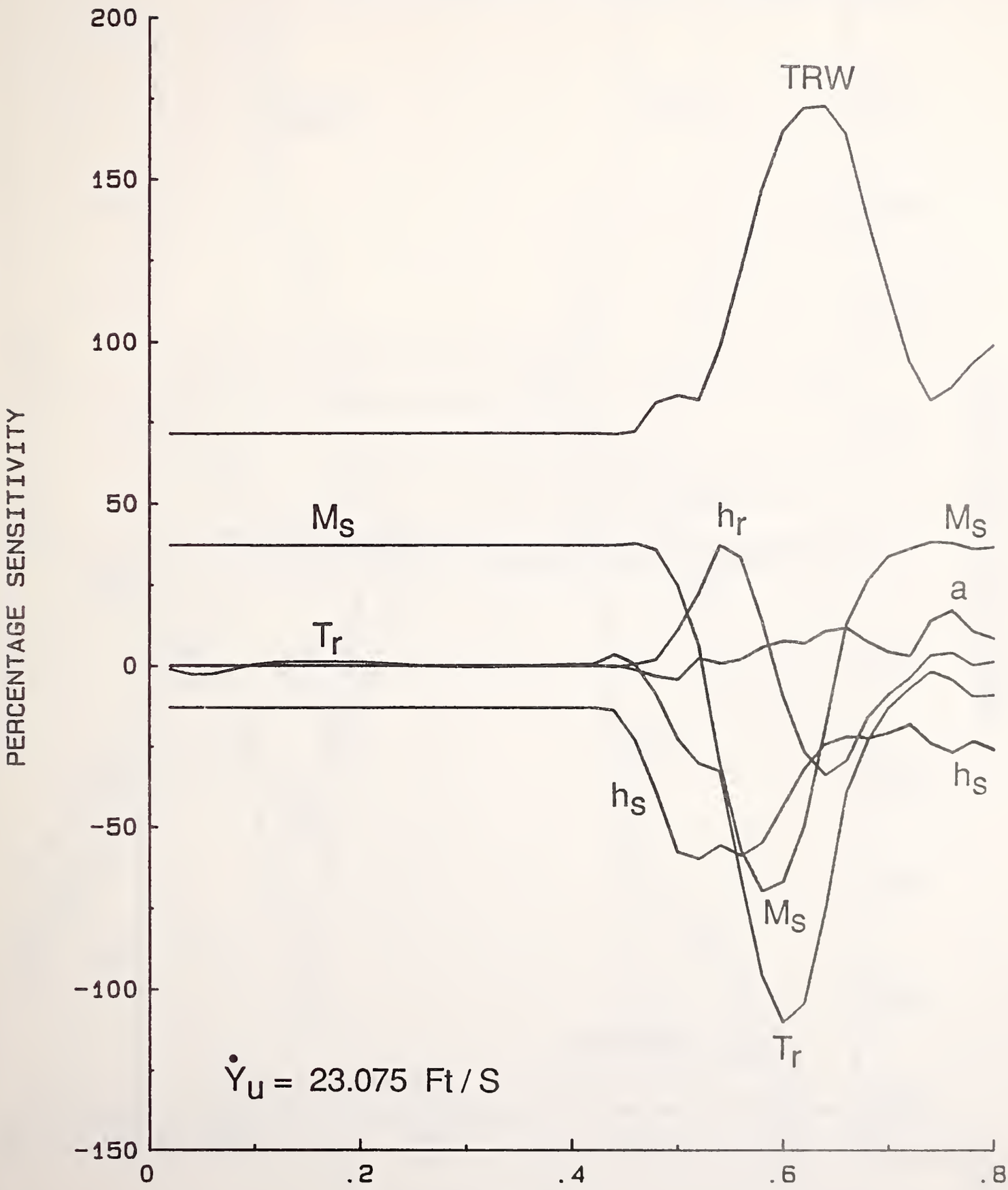
SENSITIVITY OF ROLLOVER
PREVENTION ENERGY RESERVE
200
150
100
50
0
-50
-100
-150
PERCENTAGE SENSITIVITY
TRW
Ms
hr
Ms
a
Tr
hs
hs
Ms
Tr
Ẏu = 23.075 Ft / S
0
.2
.4
.6
.8
TIME (Sec.)

Fig. 43

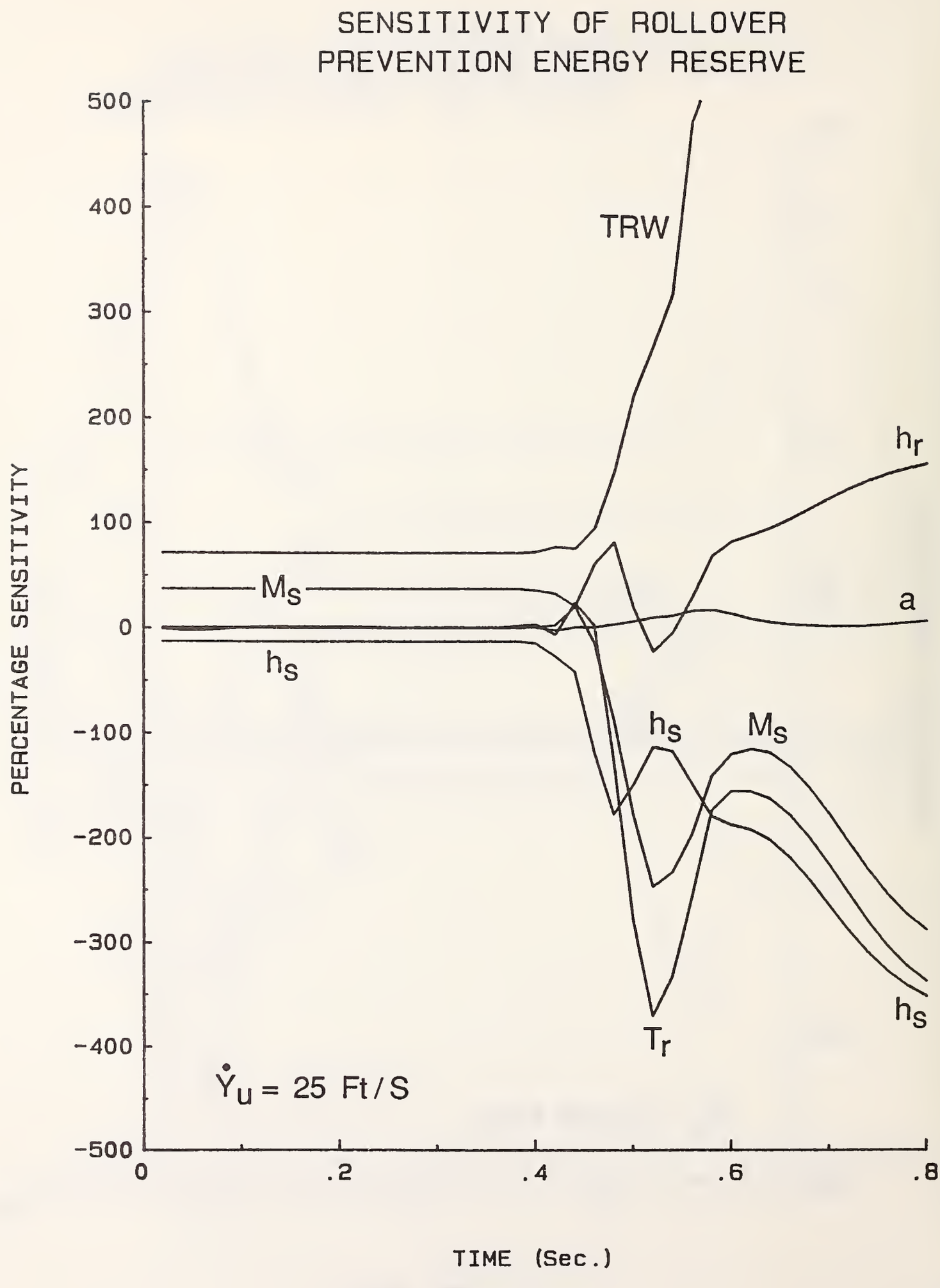
SENSITIVITY OF ROLLOVER
PREVENTION ENERGY RESERVE
PERCENTAGE SENSITIVITY
500
400
300
200
100
0
-100
-200
-300
-400
-500
0
.2
.4
.6
.8
TIME (Sec.)
TRW
h_r
M_s
a
h_s
h_s
M_s
h_s
T_r
$\dot{Y}_u$ = 25 Ft/S

Fig. 44

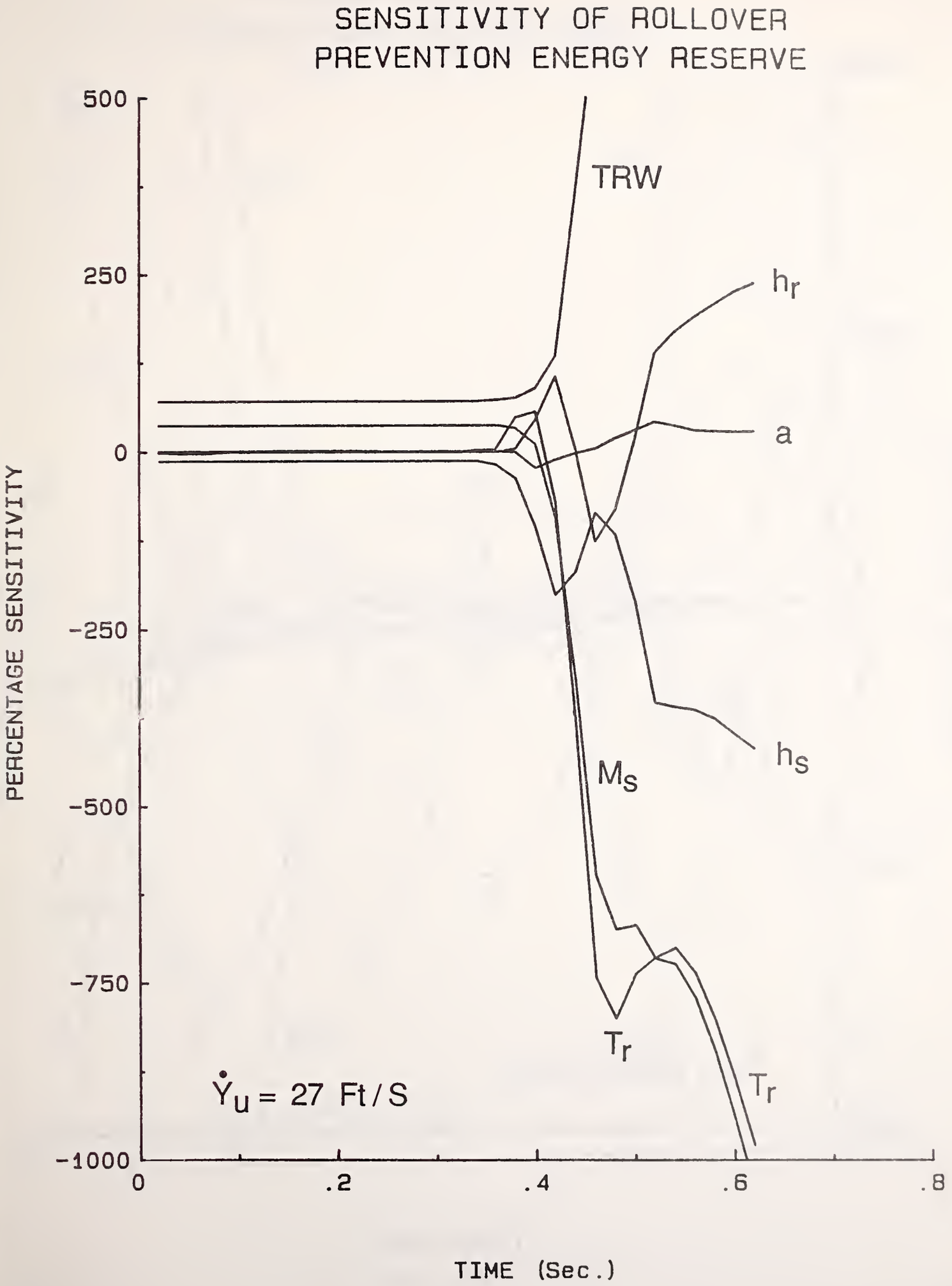
SENSITIVITY OF ROLLOVER
PREVENTION ENERGY RESERVE
500
250
0
-250
-500
-750
-1000
PERCENTAGE SENSITIVITY
TRW
h_r
a
h_s
M_s
T_r
T_r
$\dot{Y}_u$ = 27 Ft/S
0
.2
.4
.6
.8
TIME (Sec.)

Fig. 45

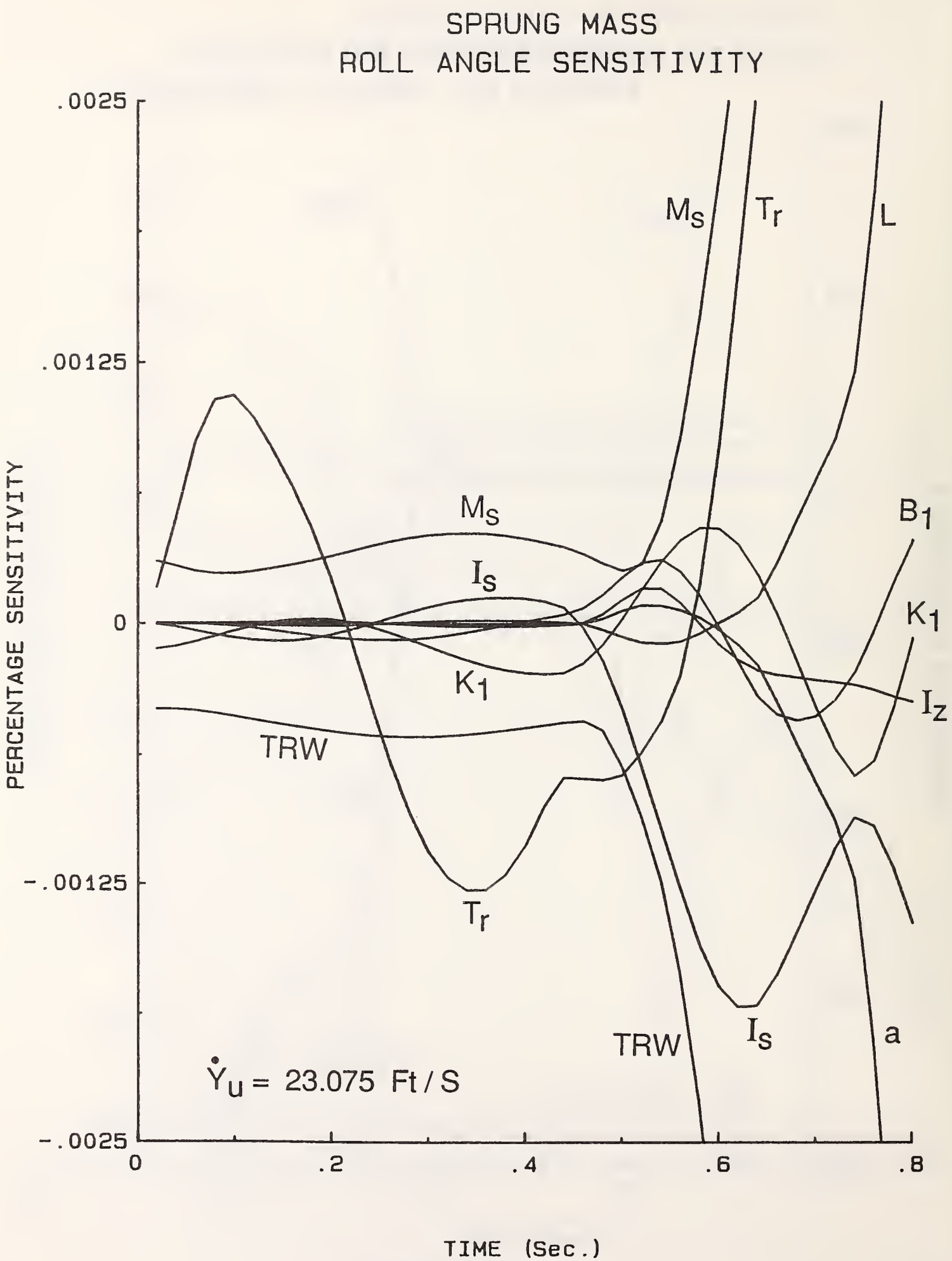
SPRUNG MASS
ROLL ANGLE SENSITIVITY
.0025
.00125
0
-.00125
-.0025
PERCENTAGE SENSITIVITY
0
.2
.4
.6
.8
TIME (Sec.)
M_S
T_r
L
M_S
I_S
K_1
TRW
T_r
B_1
K_1
I_z
TRW
I_S
a
$\dot{Y}_u$ = 23.075 Ft/S

Fig. 46

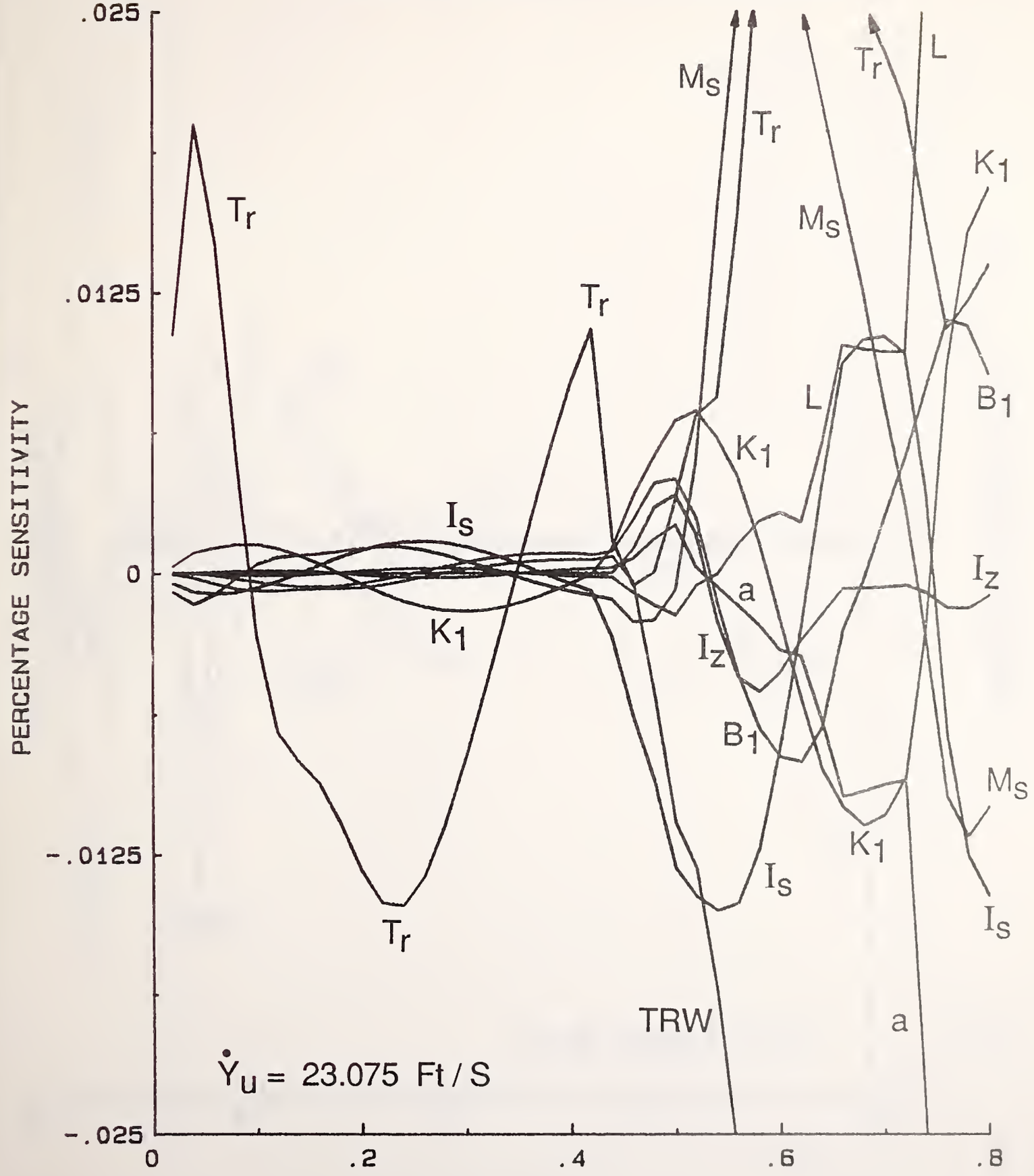
SPRUNG MASS
ROLL VELOCITY SENSITIVITY
.025
.0125
0
-.0125
-.025
PERCENTAGE SENSITIVITY
0
.2
.4
.6
.8
TIME (Sec.)
$\dot{Y}_u$ = 23.075 Ft/S
Tr
Ms
L
K1
B1
Is
Iz
a
TRW

Fig. 47

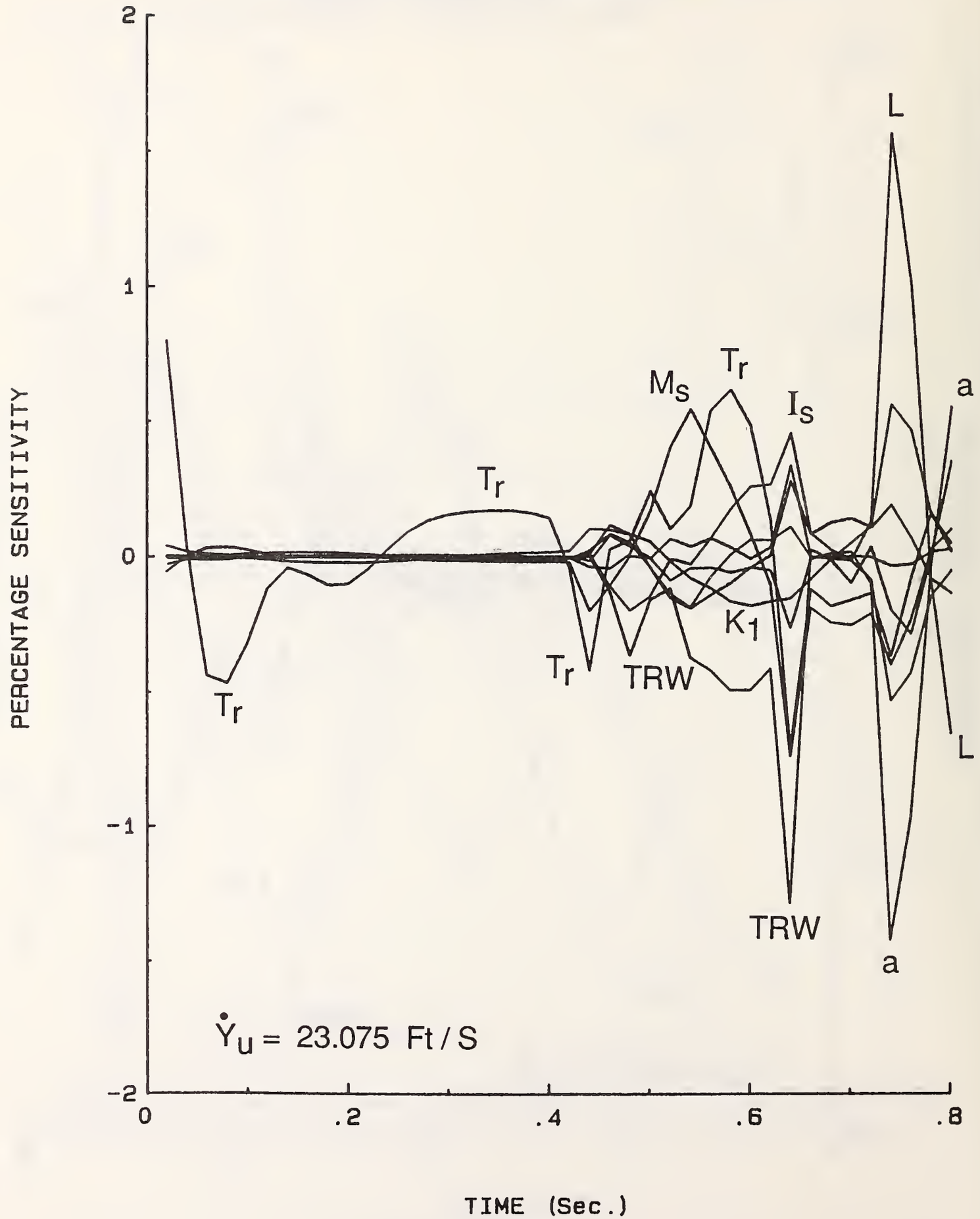
SPRUNG MASS
ROLL ACCELERATION SENSITIVITY
PERCENTAGE SENSITIVITY
2
1
0
-1
-2
0
.2
.4
.6
.8
TIME (Sec.)
L
Tr
Ms
Tr
Is
a
Tr
Tr
K1
Tr
TRW
L
TRW
a
$\dot{Y}_u$ = 23.075 Ft/S

Fig. 48

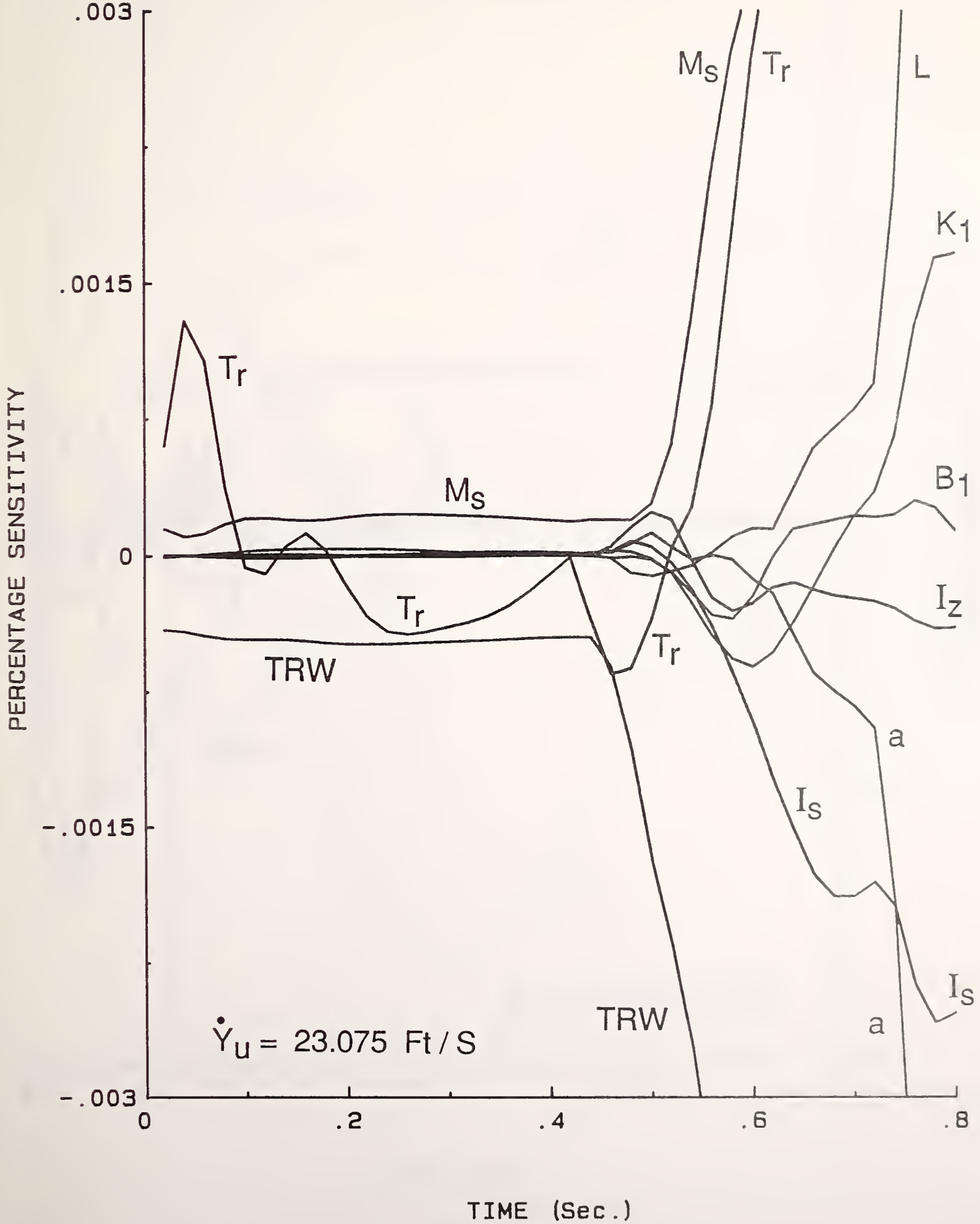
UNSPRUNG MASS
ROLL ANGLE SENSITIVITY
.003
.0015
0
-.0015
-.003
PERCENTAGE SENSITIVITY
0
.2
.4
.6
.8
TIME (Sec.)
$\dot{Y}_u$ = 23.075 Ft/S
M_S
T_r
L
K_1
B_1
I_z
a
I_S
TRW

Fig. 49

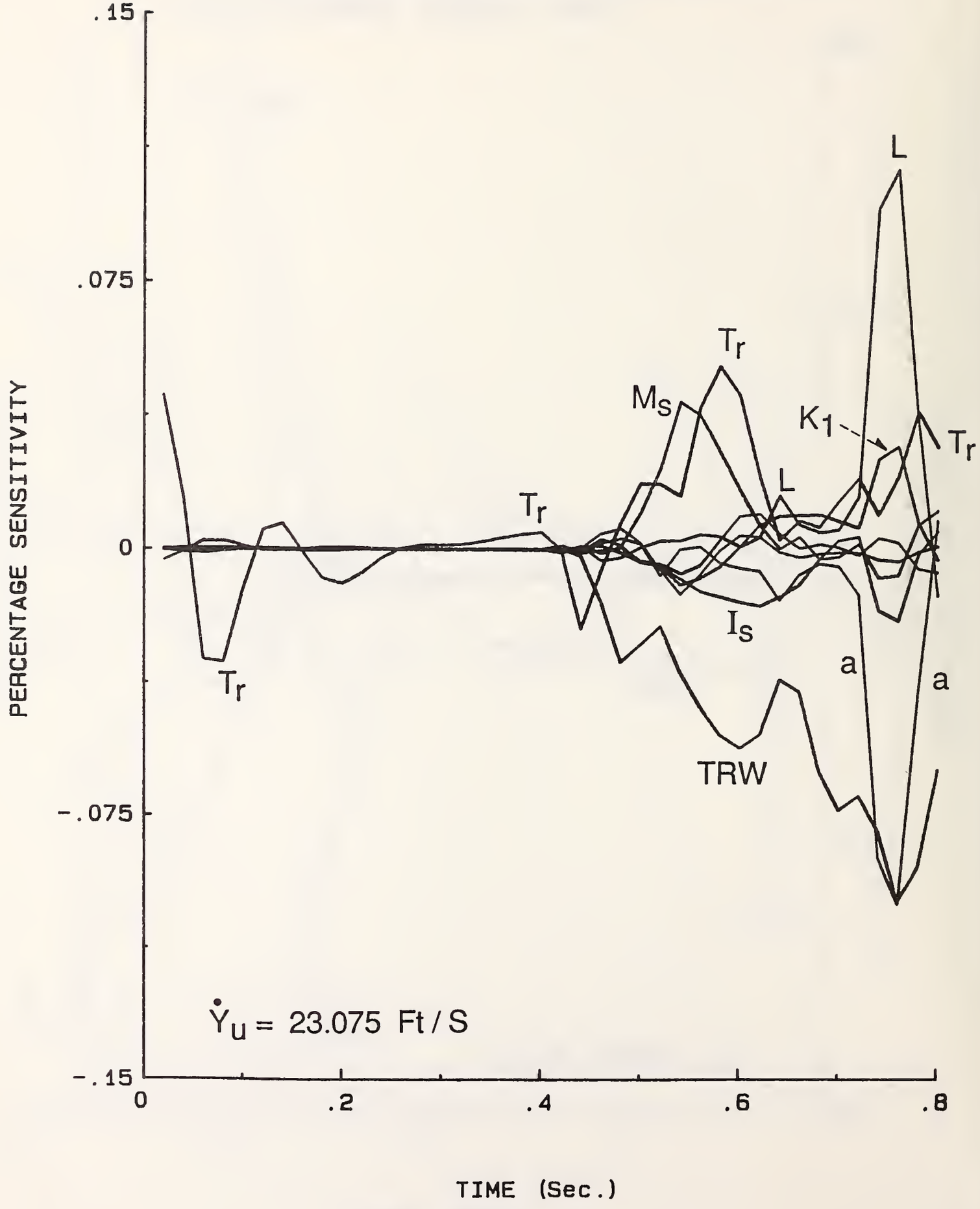
UNSPRUNG MASS
ROLL VELOCITY SENSITIVITY
.15
.075
0
-.075
-.15
PERCENTAGE SENSITIVITY
0
.2
.4
.6
.8
TIME (Sec.)
L
Tr
Ms
K1
Tr
L
Tr
Is
a
a
Tr
TRW
Ẏu = 23.075 Ft/S

Fig. 50

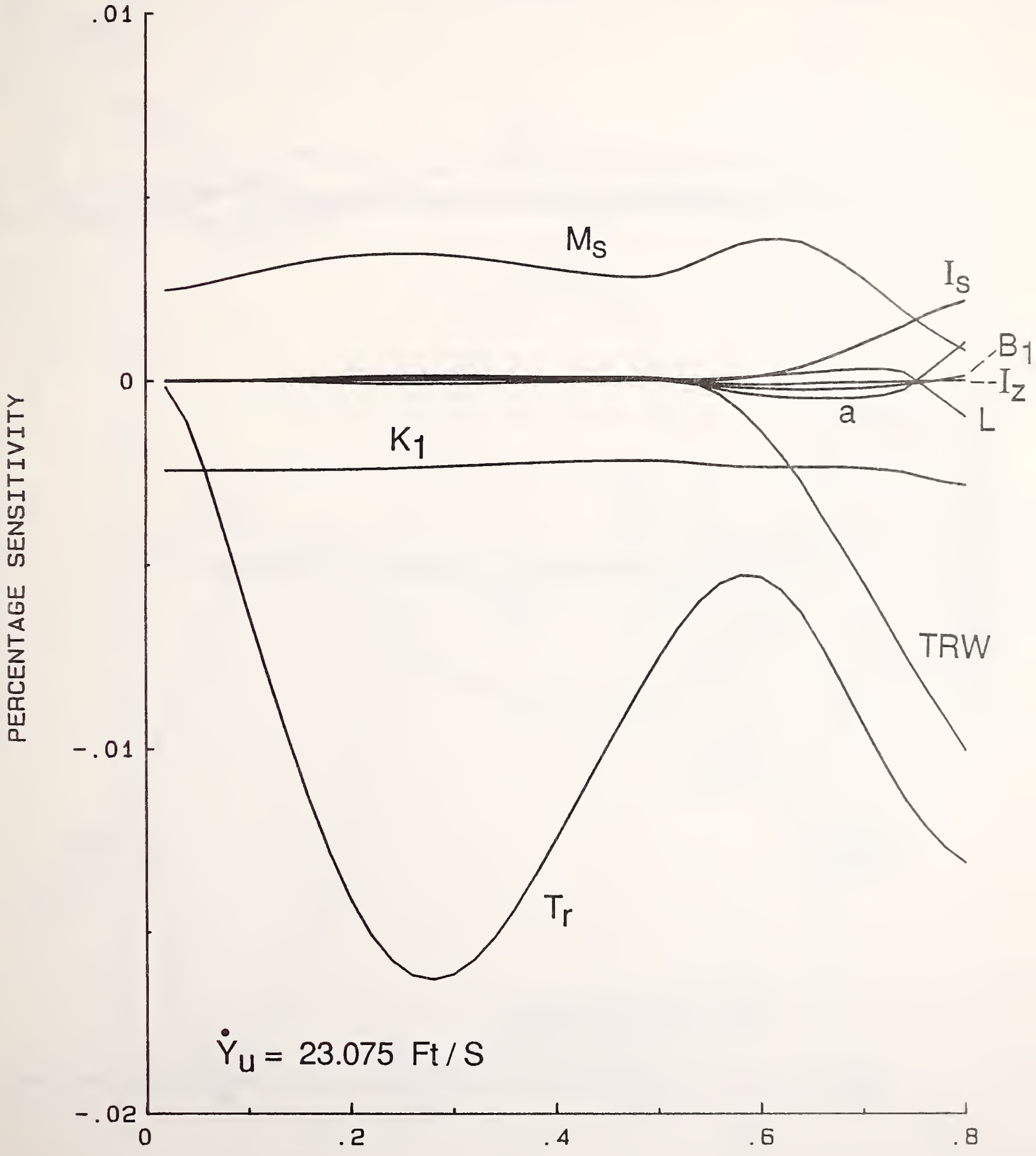
SPRUNG MASS
C.G. HEIGHT SENSTIVITY
.01
0
-.01
-.02
PERCENTAGE SENSITIVITY
M_S
I_S
B_1
I_Z
a
L
K_1
TRW
T_r
$\dot{Y}_u$ = 23.075 Ft / S
0
.2
.4
.6
.8
TIME (Sec.)

Fig. 51

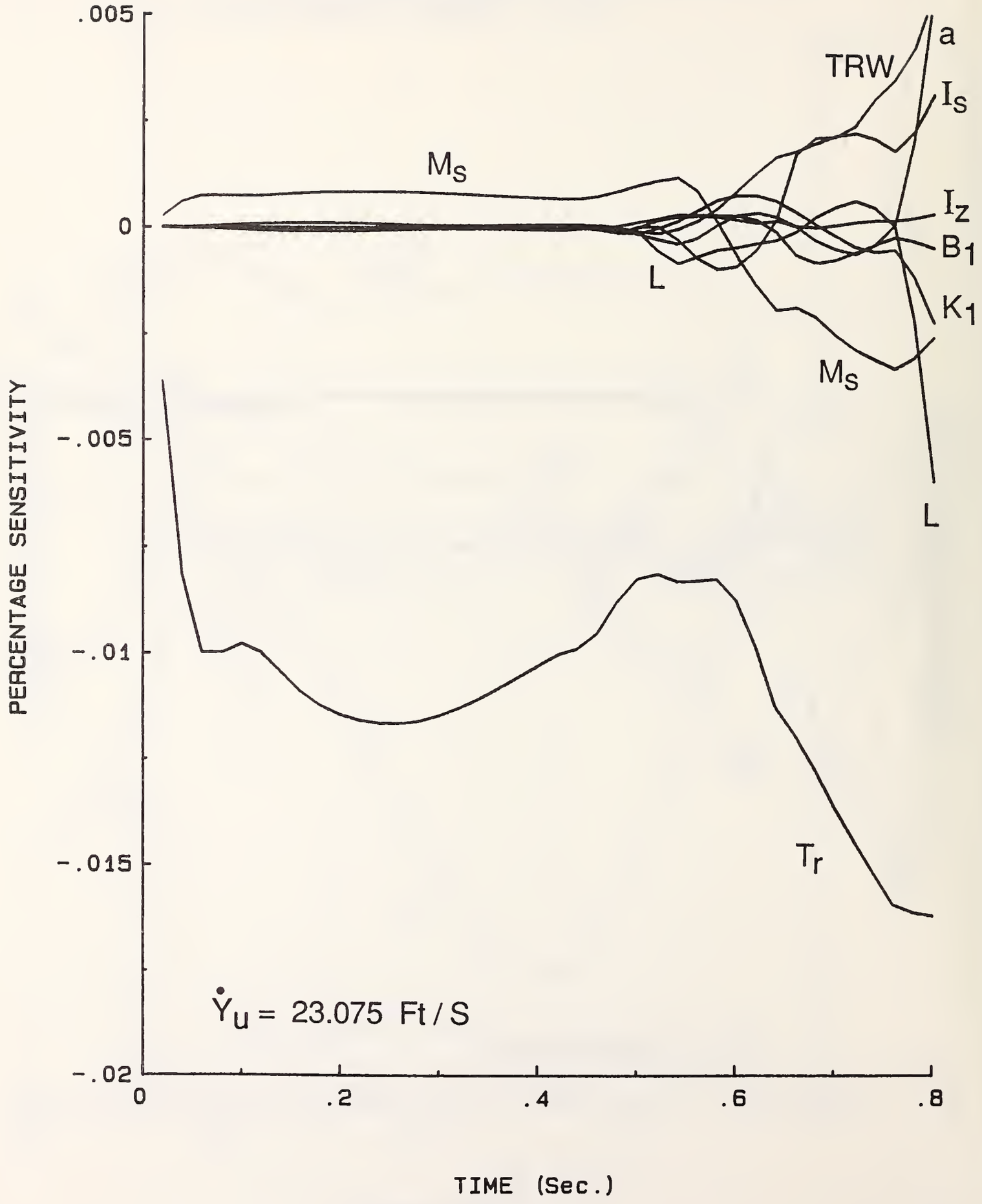
UNSPRUNG MASS
C.G. HEIGHT SENSITIVITY
.005
0
-.005
-.01
-.015
-.02
PERCENTAGE SENSITIVITY
0
.2
.4
.6
.8
TIME (Sec.)
a
TRW
I_S
M_S
I_Z
B_1
L
K_1
M_S
L
T_r
$\dot{Y}_u$ = 23.075 Ft / S

Fig. 52

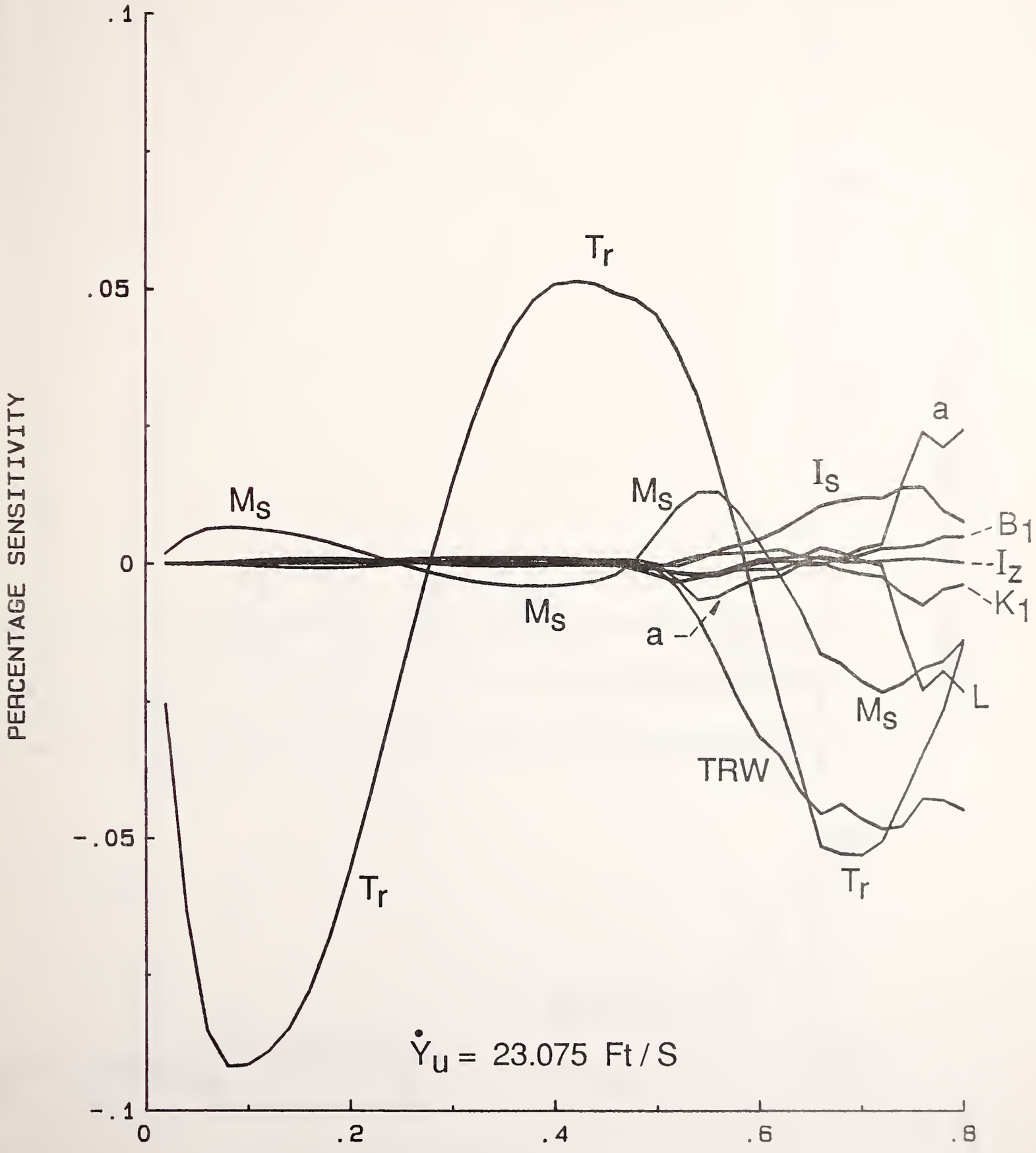
SPRUNG MASS
VERTICAL VELOCITY SENSITIVITY
PERCENTAGE SENSITIVITY
.1
.05
0
-.05
-.1
0
.2
.4
.6
.8
TIME (Sec.)
Tr
Ms
Ms
Ms
Is
a
B1
Iz
K1
L
Ms
TRW
Tr
a
Tr
Ẏu = 23.075 Ft/S

Fig. 53

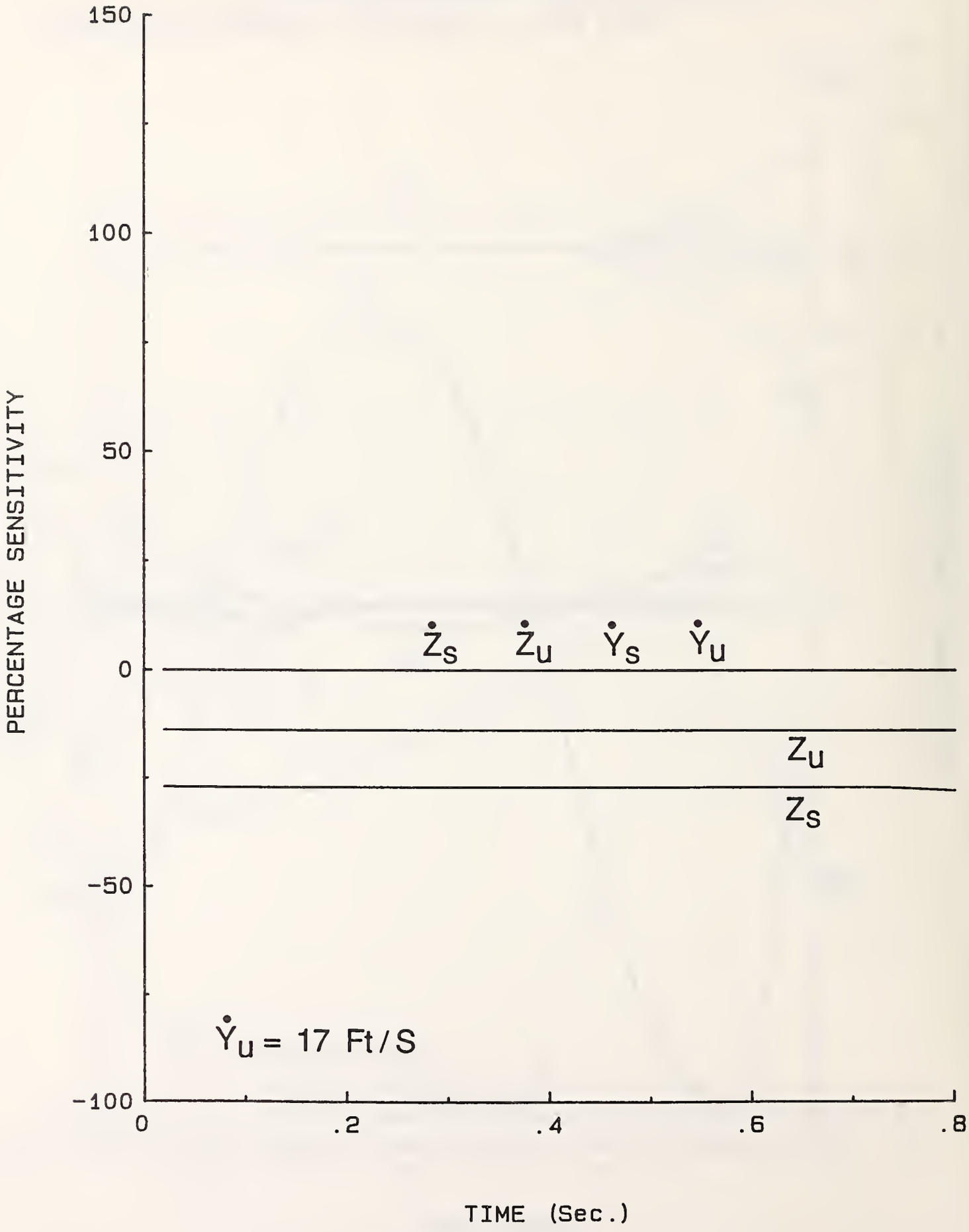
SENSITIVITY OF RP ENERGY RESERVE
SYSTEM RESPONSE PARAMETER SET
PERCENTAGE SENSITIVITY
150
100
50
0
-50
-100
0
.2
.4
.6
.8
TIME (Sec.)
$\dot{Z}_s$ $\dot{Z}_u$ $\dot{Y}_s$ $\dot{Y}_u$
Z_u
Z_s
$\dot{Y}_u$ = 17 Ft/S

Fig. 54

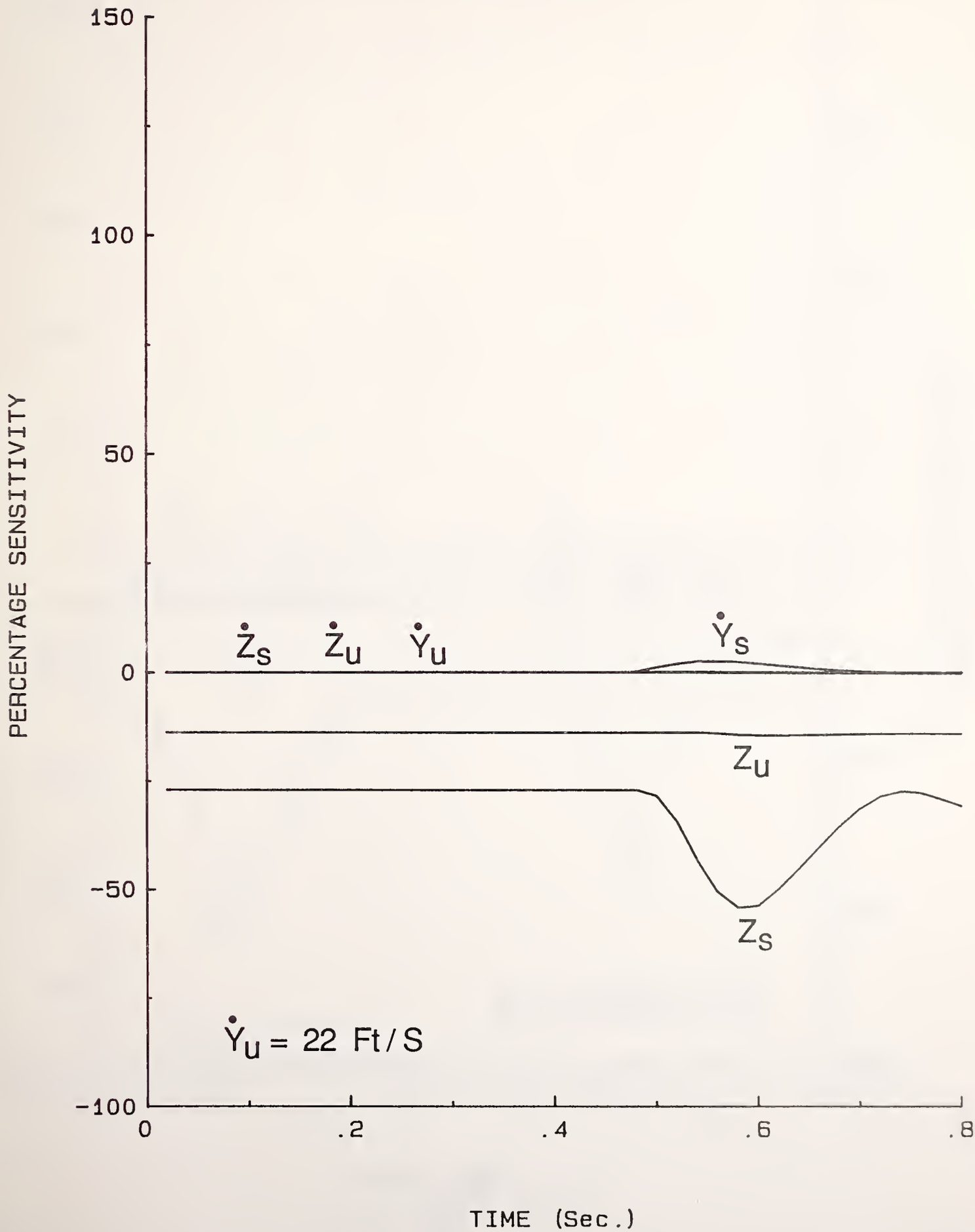
SENSITIVITY OF RP ENERGY RESERVE
SYSTEM RESPONSE PARAMETER SET
150
100
50
0
-50
-100
PERCENTAGE SENSITIVITY
$\dot{Z}_S$ $\dot{Z}_u$ $\dot{Y}_u$
$\dot{Y}_S$
Z_u
Z_S
$\dot{Y}_u$ = 22 Ft/S
0
.2
.4
.6
.8
TIME (Sec.)

Fig. 55

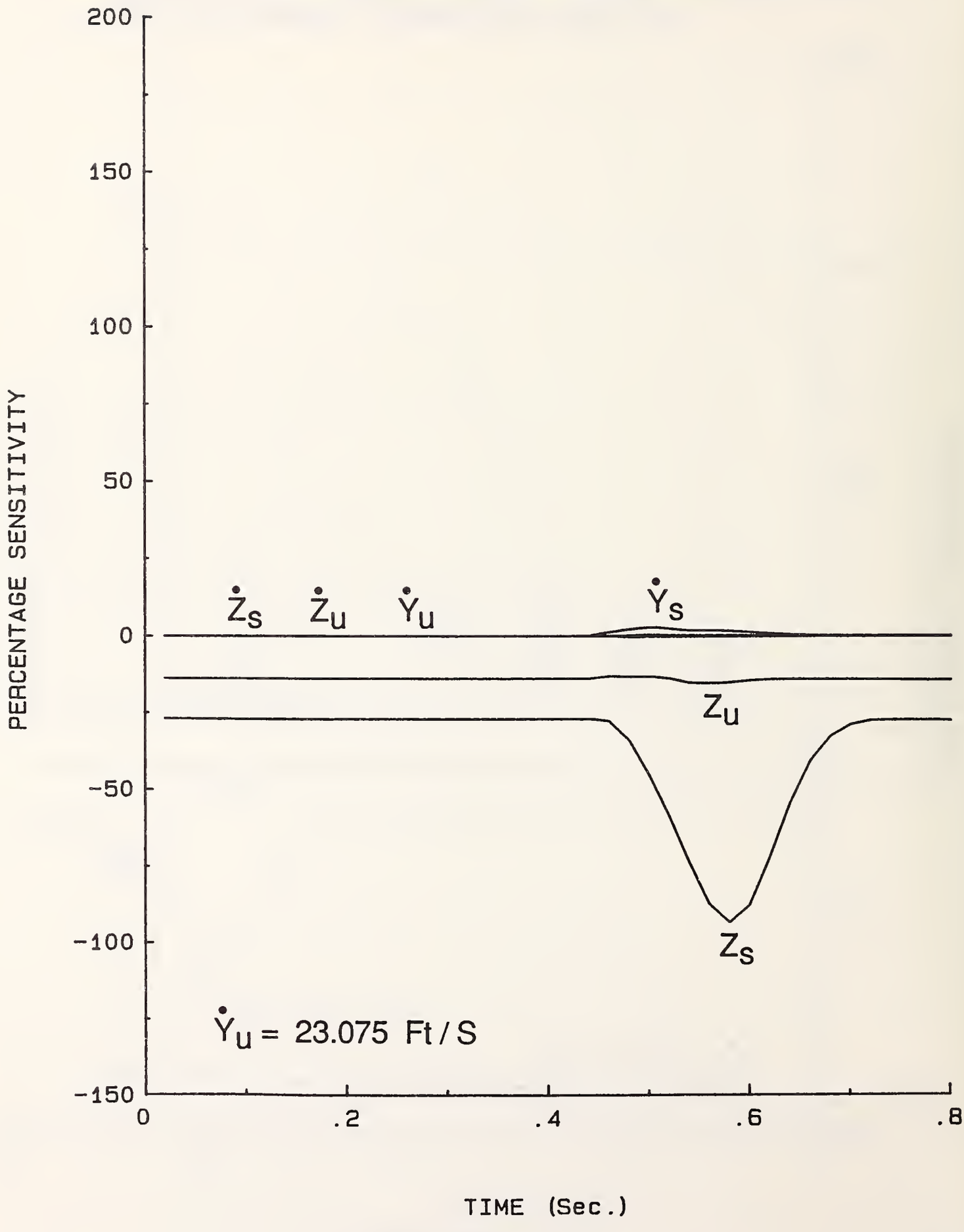
SENSITIVITY OF RP ENERGY RESERVE
SYSTEM RESPONSE PARAMETER SET
200
150
100
50
0
-50
-100
-150
PERCENTAGE SENSITIVITY
$\dot{Z}_s$ $\dot{Z}_u$ $\dot{Y}_u$
$\dot{Y}_s$
Z_u
Z_s
$\dot{Y}_u$ = 23.075 Ft / S
0
.2
.4
.6
.8
TIME (Sec.)

Fig. 56

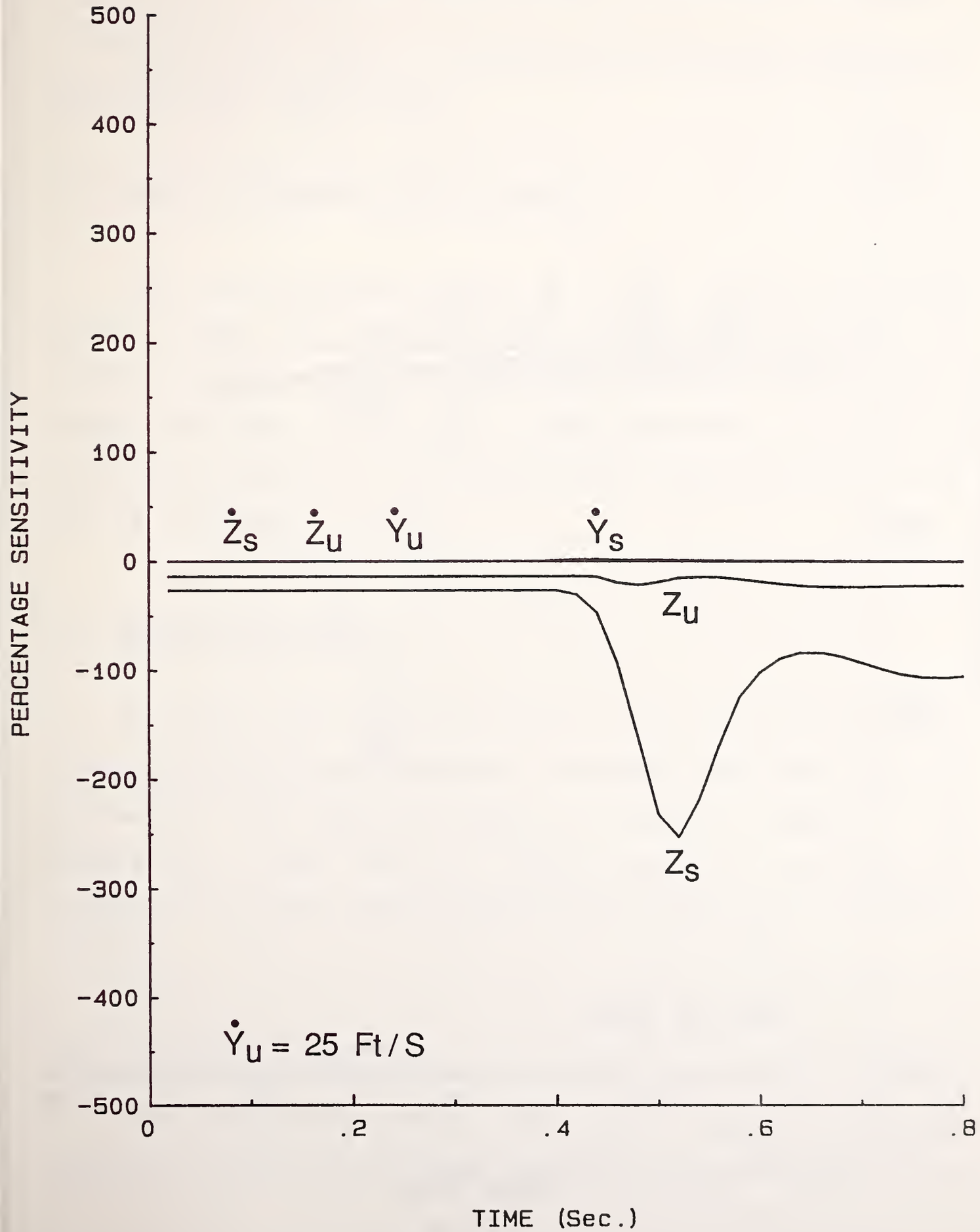
SENSITIVITY OF RP ENERGY RESERVE
SYSTEM RESPONSE PARAMETER SET
PERCENTAGE SENSITIVITY
500
400
300
200
100
0
-100
-200
-300
-400
-500
0
.2
.4
.6
.8
TIME (Sec.)
$\dot{Z}_S$
$\dot{Z}_u$
$\dot{Y}_u$
$\dot{Y}_S$
Z_u
Z_S
$\dot{Y}_u = 25$ Ft/S

Fig. 57

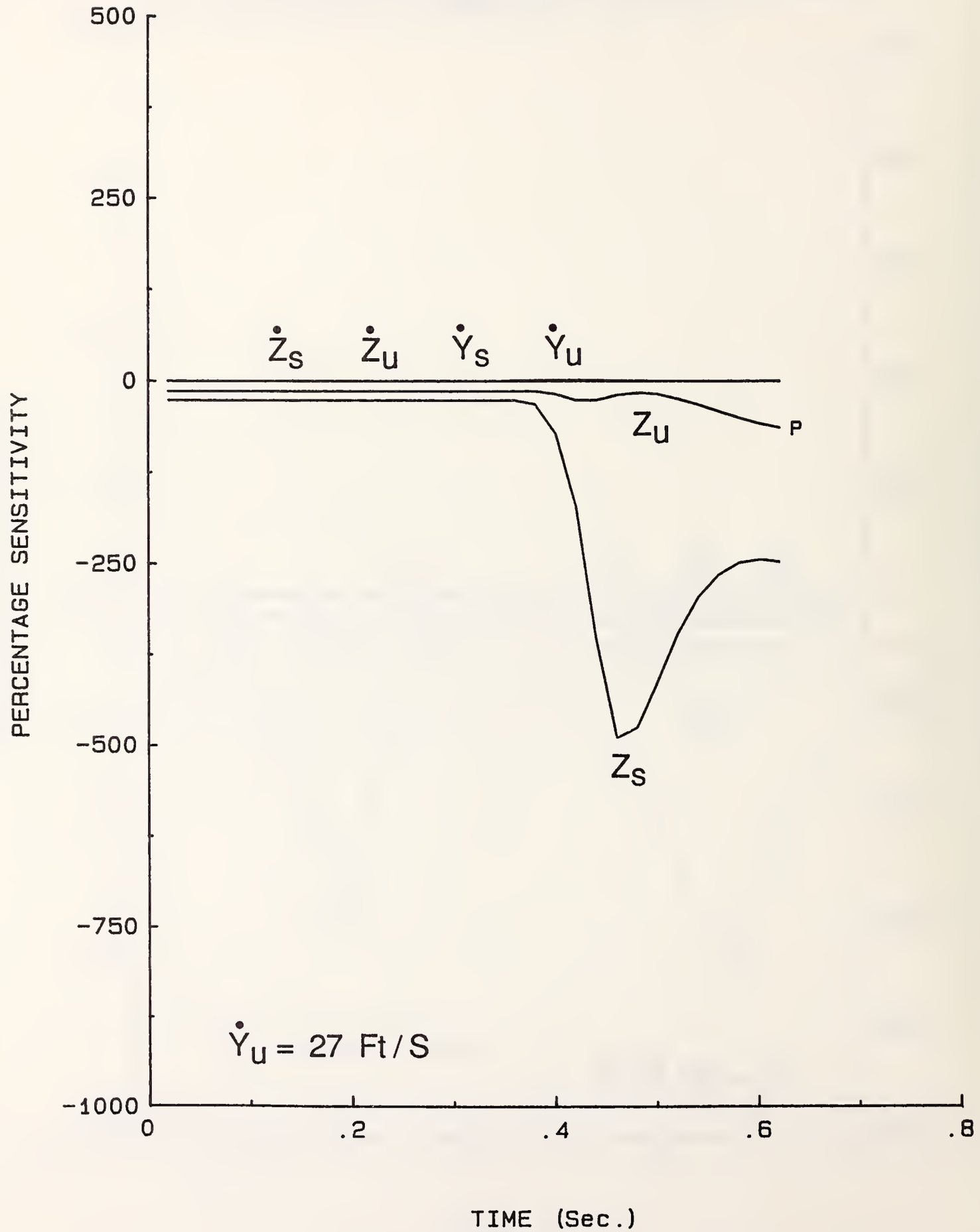
SENSITIVITY OF RP ENERGY RESERVE
SYSTEM RESPONSE PARAMETER SET
500
250
0
-250
-500
-750
-1000
PERCENTAGE SENSITIVITY
$\dot{Z}_S$ $\dot{Z}_u$ $\dot{Y}_S$ $\dot{Y}_u$
Z_u
P
Z_S
$\dot{Y}_u$ = 27 Ft/S
0
.2
.4
.6
.8
TIME (Sec.)

Fig. 58

7.0 CONCLUSIONS AND RECOMMENDATIONS

Based on the results of the sensitivity analysis the following conclusions were reached.

7.1 Rollover Prevention Energy Reserve

The stability based concept of rollover prevention energy reserve (RPER) is a useful tool for examining the influence of vehicle parameters in tripped rollover maneuvers. The results reveal that the minimum value of RPER approaches zero at the threshold of rollover, and if the minimum falls below zero then vehicle rollover will occur.

7.2 Sensitivity Results

In general, the results obtained using the percentage sensitivity functions are reasonable and agree with the existing information and opinions regarding rollover behavior and propensity. While this in itself does not validate the sensitivity methods used in this report it certainly lends them credence.

7.2.1 Geometric Parameters

The sensitivity results show that vehicle geometric parameters are by far the most influential characteristics in a

tripped rollover situation. The results indicate that, in tripped rollover maneuvers, vehicle stability can be best enhanced by increasing vehicle track width or reducing vehicle c.g. height. It was found that geometric parameters commonly associated with suspension components, such as the distance separating the suspension springs (s), and the distance from roll axis to vehicle axle (h_r), exhibited very little influence on vehicle RPER.

7.2.2 Mass Parameters

The results show that parameters generally associated with the vehicle's mass properties are relatively important, though not nearly as important as parameters associated with vehicle geometry.

The sensitivity of RPER to variations in mass parameters indicate that the stability can be increased by increasing the vehicle sprung mass (M_s). This result may be partially due to the fact that the a heavier sprung mass compresses the suspension springs, thus lowering the c.g. position of vehicle. The results also reveal that increasing roll moments of inertia tends to increase RPER by opposing vehicle motion.

7.2.3 Stiffness Parameters

The influence of vehicle stiffness parameters on vehicle RPER was found to be quite small, indicating that suspension

spring stiffness has little influence on vehicle motion in tripped rollover situations.

7.2.4 Damping and Force Parameters

Damping parameters such as the damping coefficients of shock absorbers were found to have very little influence on vehicle stability in tripped rollover maneuvers.

Parameters associated with external forces acting on the vehicle such as deformation forces caused by impact, and the points of discontinuity in the force/deformation relationship were found to have a significant influence on vehicle stability. By spreading the impact deformation over a large distance (thus increasing the amount of vehicle plastic deformation) the amount of unrecoverable work used to crush the vehicle is increased, reducing the amount of energy which can contribute to the vehicle's rolling motion.

7.3 Recommendations

Accident statistics have shown that vehicle rollover represents a serious safety problem on the nations roads and highways and that utility vehicles and small pickup trucks have the highest probability of being involved in a rollover accident. Since the sales of these vehicles have grown substantially in the past few years it is likely that the number of fatalities due to rollover accidents will increase as well. Because of these trends

there has been a growing call for the government to develop legislation which will establish minimum stability standards for passenger vehicles. If legislation aimed at curbing the number of rollover accidents is to be effective it is imperative that the influence of design parameters on vehicle rollover propensity be thoroughly understood. With this in mind we feel that more research into vehicle rollover behavior is necessary and should be directed toward the following areas.

7.3.1 Continued Investigation Into Tripped Rollover

This report presents a thorough investigation into the effects of vehicle design parameters on tripped rollover maneuvers involving simultaneous impact of front and rear wheels. It should be noted, however that the results presented in this paper were obtained using a single set of vehicle data and in one particular type of rollover accident. Because of the this we feel that the sensitivity methods used in this report should be applied to investigate several other sets of vehicle data and in different rollover situations using an improved tripped rollover vehicle model. The vehicle data sets should reflect the different classes of vehicles which are currently in use. These classes should include small and subcompact passenger vehicles, small and standard pickup trucks, and utility vehicles. Maneuvers to be investigated should include simultaneous front and rear wheel impact as well as vehicle oblique impact with the curb. An improved vehicle model should be developed which would account for all dynamic couplings in the equations of motion and

allow for a thorough accounting of energy transfer and its dissipation.

7.3.2 Investigation Into Untripped Rollover Behavior

A great deal of the controversy surrounding the rollover issue involves the apparent propensity of certain utility vehicles to rollover in collision avoidance maneuvers without the help of a tripping mechanism. The influence of vehicle design parameters on rollover propensity in these types of untripped maneuvers may differ considerably from those associated with tripped rollover. While this report found that the influence of parameters associated with suspension characteristics in tripped rollovers were small these parameters may prove to have a substantial effect on vehicle behavior in untripped maneuvers. Because of this it would be worthwhile to develop a vehicle model which would be capable of predicting the occurrence untripped rollover of vehicles in collision avoidance maneuvers. Once this is accomplished a sensitivity analysis of the model should be performed to determine the influence of vehicle design parameters on rollover propensity in untripped situations.

REFERENCES

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3. Nalecz, A. G., Wicher, J. (1988) "Design Sensitivity Analysis of Mechanical Systems in Frequency Domain", Journal of Sound and Vibration., Vol. 120, No. 3, pp. 517 - 526.

4. Wicher, J., Nalecz, A. G. (1987) "Second Order Sensitivity Analysis of Lumped Mechanical Systems in the Frequency Domain", International Journal for Numerical Methods in Engineering, Vol. 24, No.12, pp. 2357 - 2366.

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7. Nalecz, A. G., Bindemann, A. C. (1987) "Sensitivity Analysis of Vehicle Design Attributes that Affect Vehicle Response in Critical Accident Situations - Part I: User's Manual", Final Report NHTSA - U.S. DOT No. DOT HS 807229.

8. Nalecz, A. G., Bindemann, A. C. (1987) "Sensitivity Analysis of Vehicle Design Attributes that Affect Vehicle Response in Critical Accident Situations - Part II: Technical Report", Final Report NHTSA - U.S. DOT No. DOT HS 807230.

9. Nalecz, A. G., Bindemann, A. C. (1988) "Investigation Into the Stability of Four Wheel Steering Vehicles", International Journal of Vehicle Design, Vol. 9, No. 2, pp. 159 - 178.

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APPENDIX A

Vehicle Data for 1975 Chevy Nova

a	4.45	ft
L	9.25	ft
TRW	5.00	ft
h_u	0.2	ft
K_z	12000.00	lb/ft
B_z	80.00	lb-sec/ft
μ	0.75	--
B_d	0.25	lb-sec/ft
KD1	24000.00	lb/ft
KD2	96000.00	lb/ft
M_u	16.7	lb-s^2/ft
h_r	1.0	ft
K_1	6900.00	lb/ft
K_2	33000.00	lb/ft
B_1	325.00	lb-sec/ft
B_2	33.00	lb-sec/ft
I_x	330.00	lb-ft/s^2
I_Y	2240.00	lb-ft/s^2
I_Z	2240.00	lb-ft/s^2
h_s	1.2	ft
I_u	132.00	lb-ft/s^2
WHLDEF	4000.00	lb
K_v	1.00	ft/s
POINT1	0.1667	ft

Vehicle Data for 1975 Chevy Nova (Concluded)

POINT2	0.6667	ft
M_s	105.00	lb-s^2/ft
I_s	240.00	lb-ft/s^2
T_r	1.0	ft
H_{CURB}	0.5	ft
s	1.75	ft
K	200000.00	lb/ft
SPRLNG	0.6667	--
ABSDEF	0.25	ft
B	1000.00	lb-s/ft

APPENDIX B

Initial Conditions for Tripped Rollover Maneuvers

Y_u dot	varied*	ft/s
Z_u dot	0.00	ft/s
Phi_u dot	0.00	rad/s
Y_s dot	varied*	ft/s
Z_s dot	0.00	ft/s
Phi_s dot	0.00	rad/s
Theta dot	0.00	rad/s
Psi dot	0.00	rad/s
Y_u	10.00**	ft
Z_u	-1.11	ft
Phi_u	1.1568	rad
Y_s	10.00**	ft
Z_s	-2.5319	ft
Phi_s	1.8437	rad
Theta	0.00	rad
Psi	0.00	rad

* Initial lateral speeds used were 17 ft/s, 22 ft/s, 23.075 ft/s, 25 ft/s, and 27 ft/s

** Vehicle initial position was 7.5 ft measured from the front right tire, the location of the added non-inertial reference system.

APPENDIX C

System Response of Vehicle in Tripped Rollover

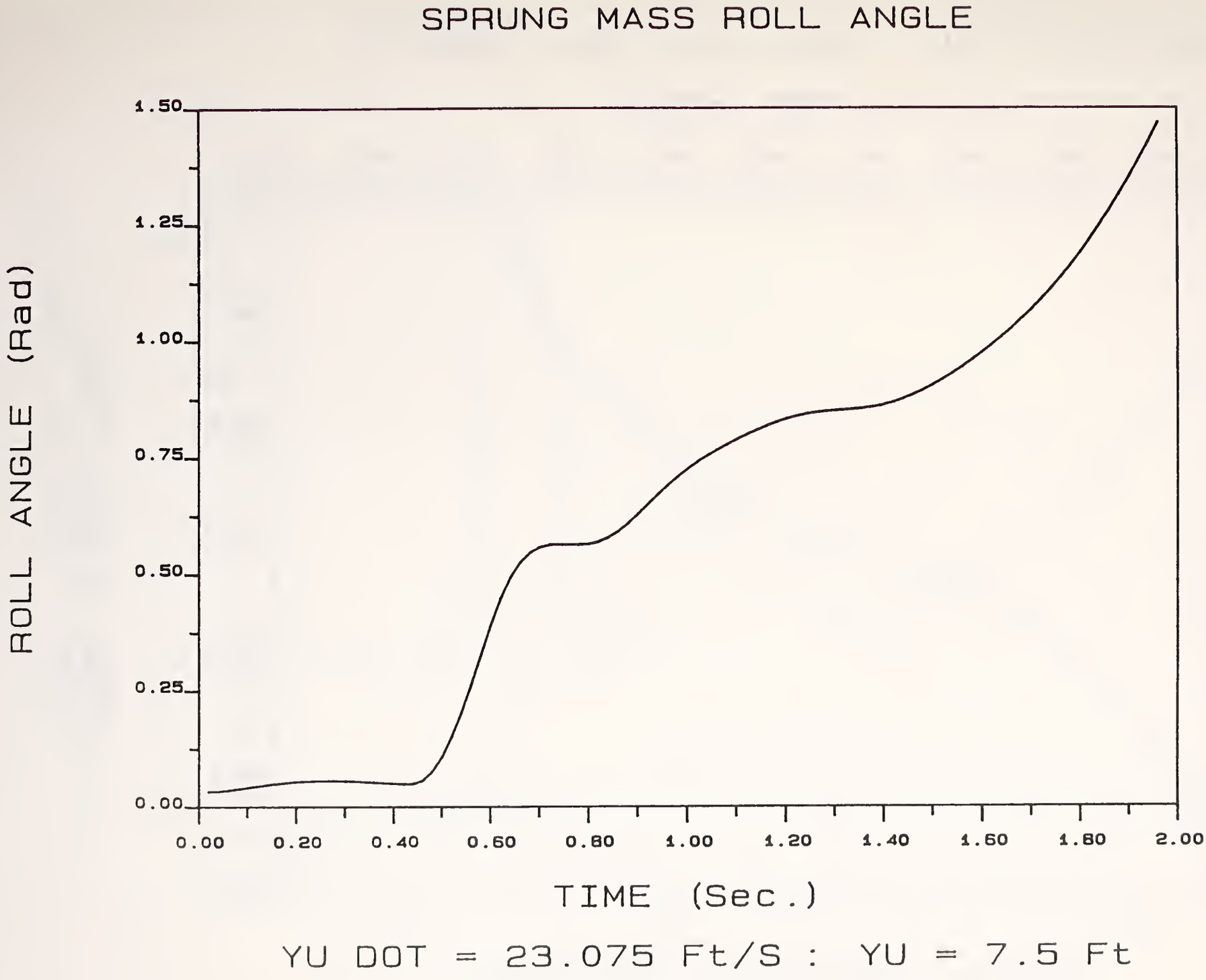
SPRUNG MASS ROLL ANGLE
ROLL ANGLE (Rad)
1.50
1.25
1.00
0.75
0.50
0.25
0.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec.)
YU DOT = 23.075 Ft/S : YU = 7.5 Ft

Fig. C-1

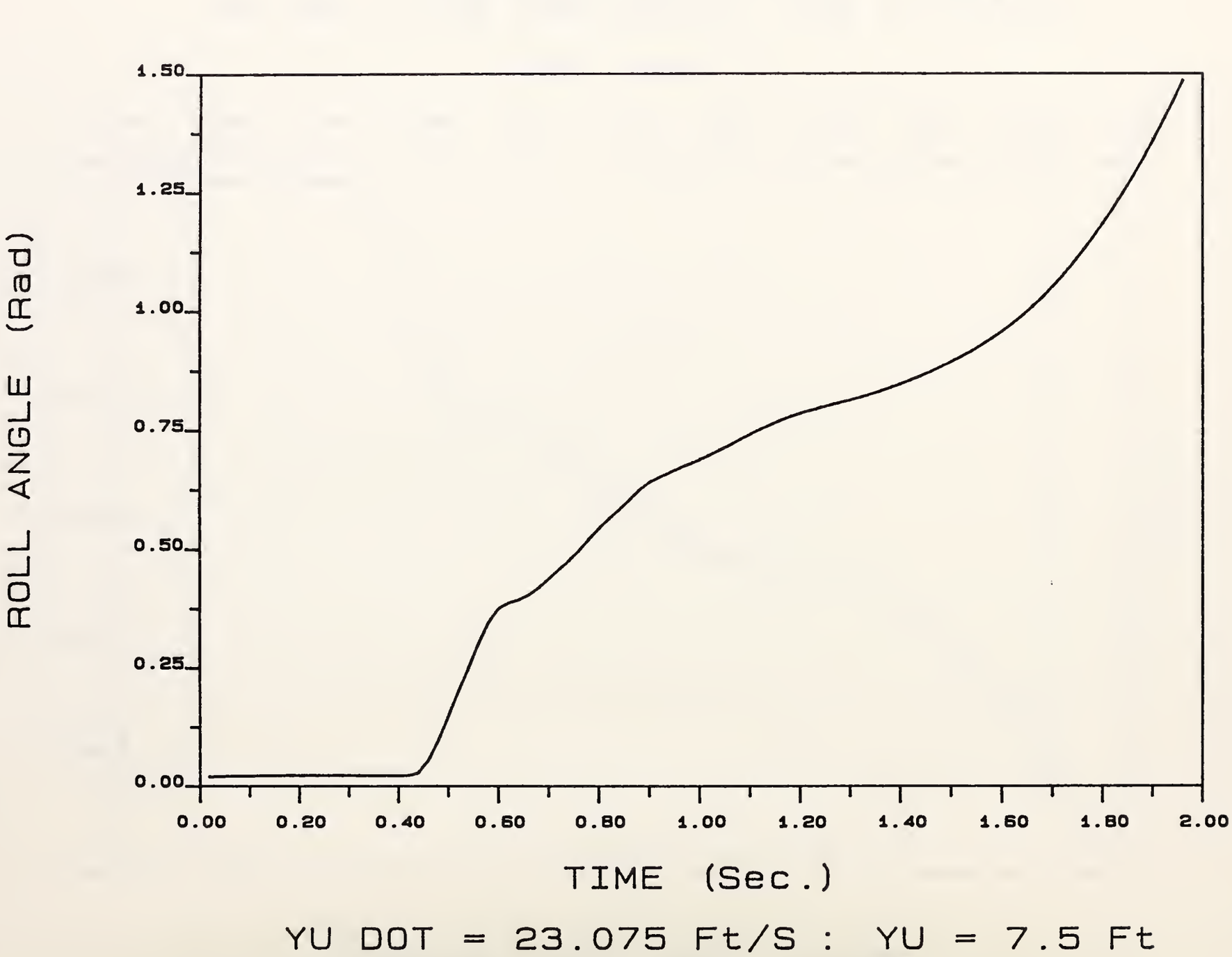
UNSPRUNG MASS ROLL ANGLE
ROLL ANGLE (Rad)
1.50
1.25
1.00
0.75
0.50
0.25
0.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec.)
YU DOT = 23.075 Ft/S : YU = 7.5 Ft

Fig. C-2

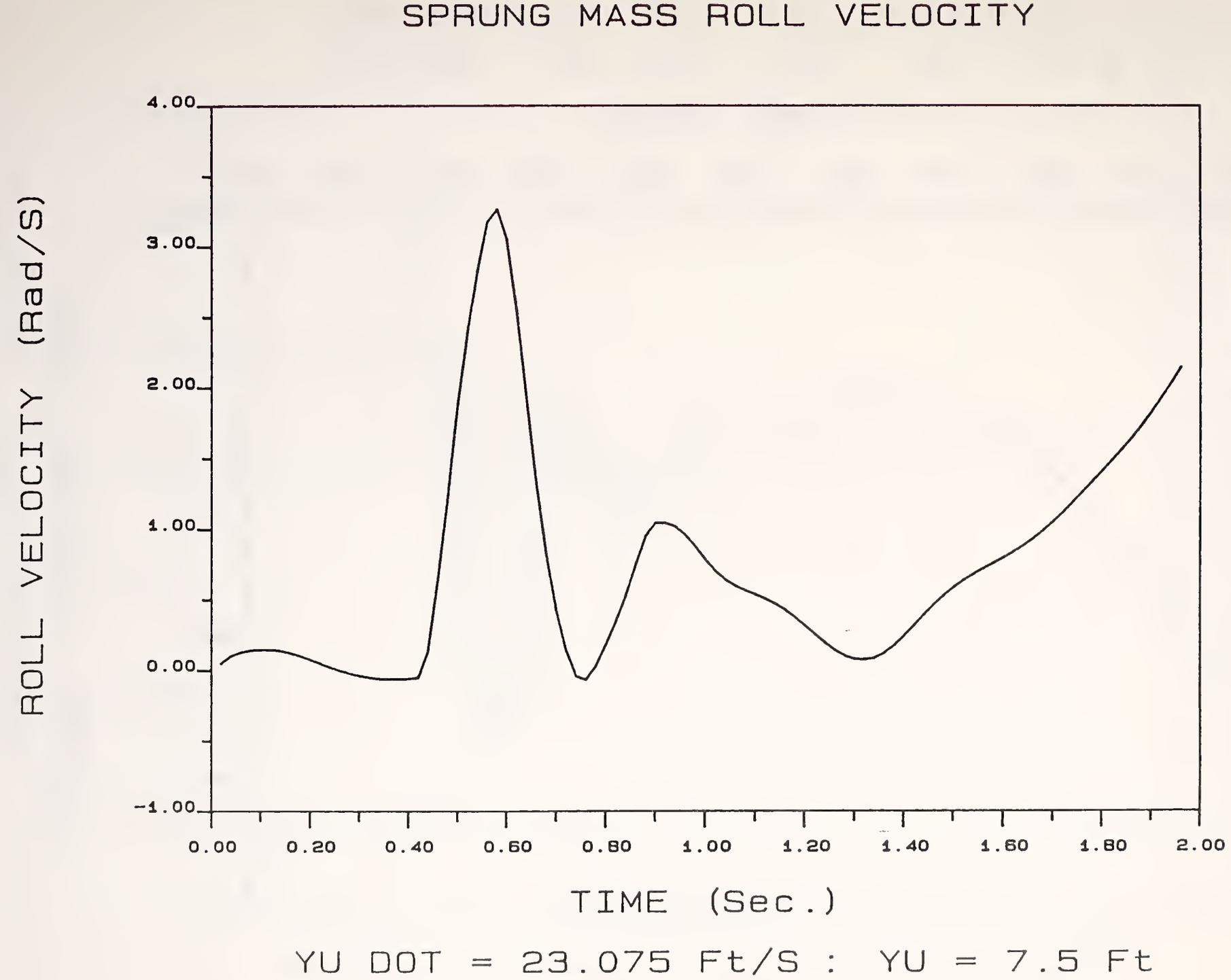
SPRUNG MASS ROLL VELOCITY
ROLL VELOCITY (Rad/S)
4.00
3.00
2.00
1.00
0.00
-1.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec.)
YU DOT = 23.075 Ft/S : YU = 7.5 Ft

Fig. C-3

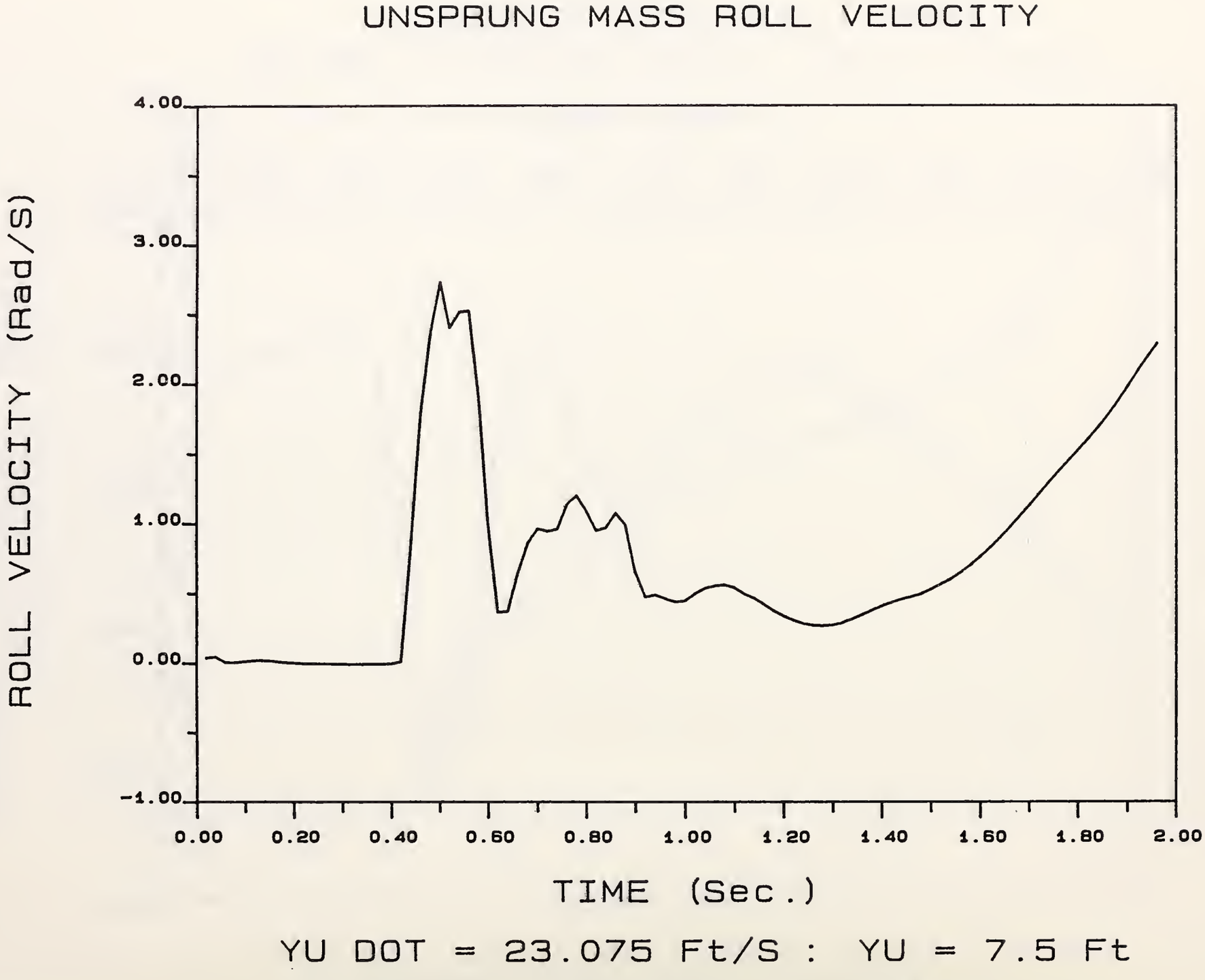
UNSPRUNG MASS ROLL VELOCITY
ROLL VELOCITY (Rad/S)
4.00
3.00
2.00
1.00
0.00
-1.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec.)
YU DOT = 23.075 Ft/S : YU = 7.5 Ft

Fig. C-4

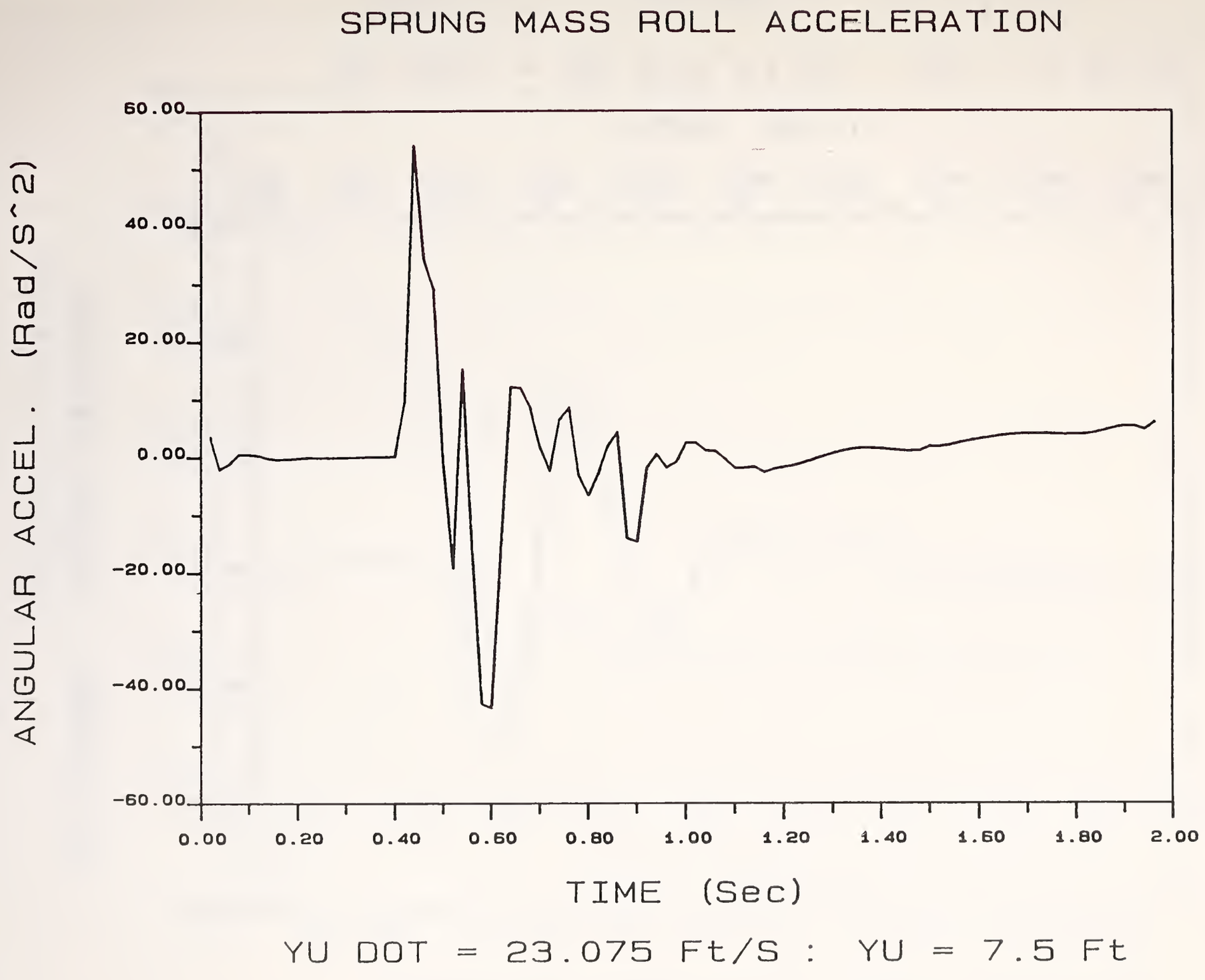
SPRUNG MASS ROLL ACCELERATION
ANGULAR ACCEL. (Rad/S^2)
60.00
40.00
20.00
0.00
-20.00
-40.00
-60.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YU = 7.5 Ft

Fig. C-5

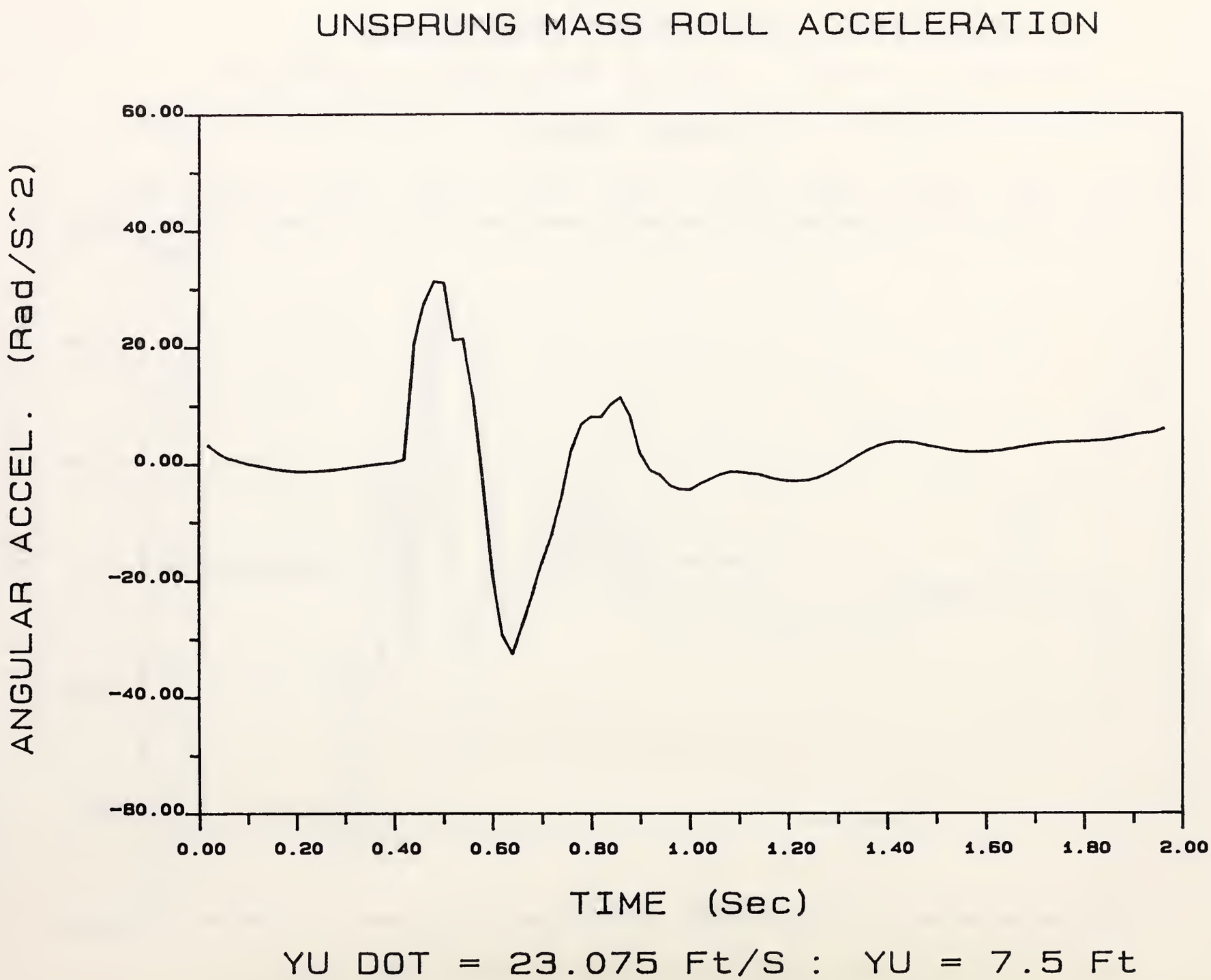
UNSPRUNG MASS ROLL ACCELERATION
ANGULAR ACCEL. (Rad/S^2)
60.00
40.00
20.00
0.00
-20.00
-40.00
-80.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YU = 7.5 Ft

Fig. C-6

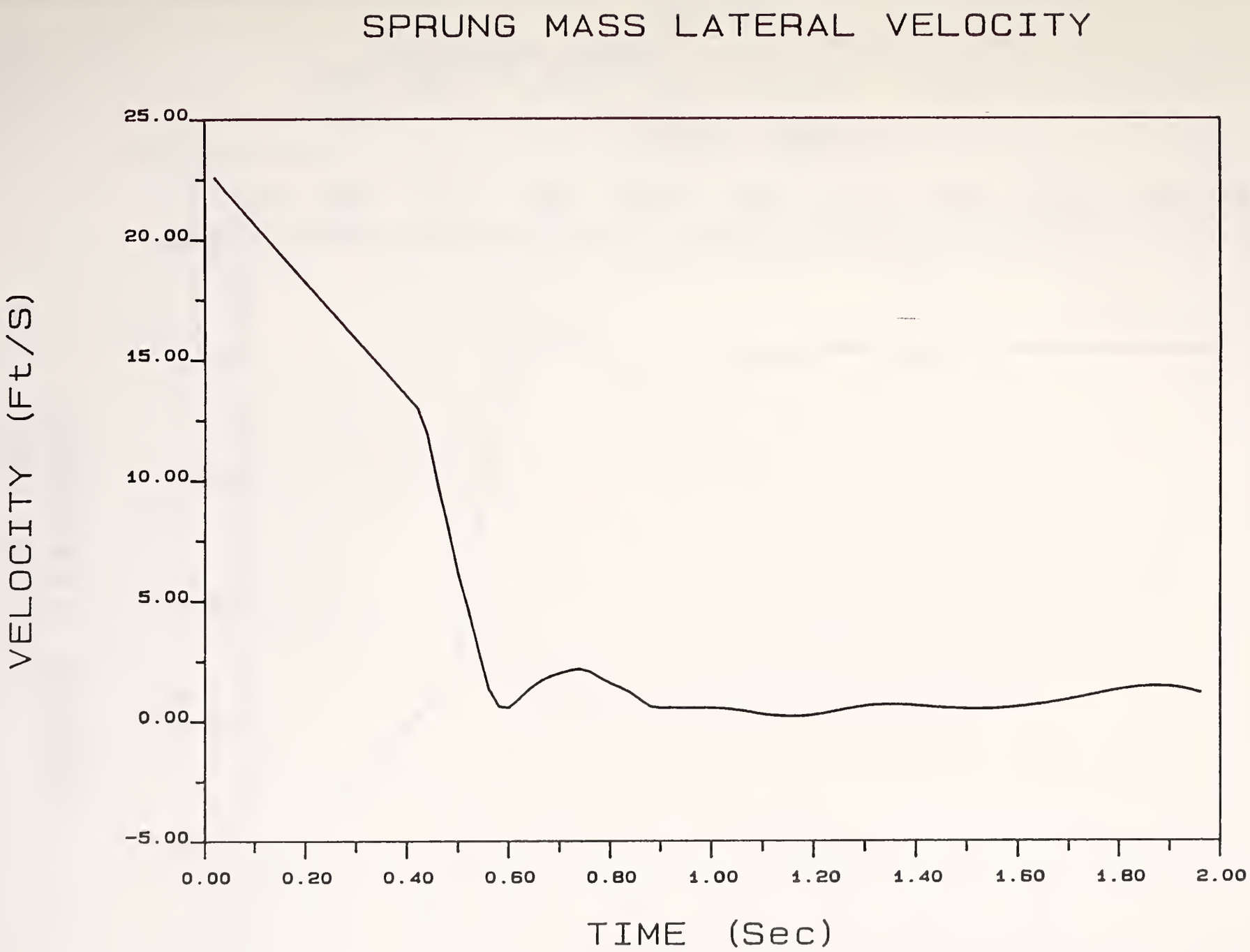
SPRUNG MASS LATERAL VELOCITY
VELOCITY (Ft/S)
25.00
20.00
15.00
10.00
5.00
0.00
-5.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YO = 7.5 Ft.

Fig. C-7

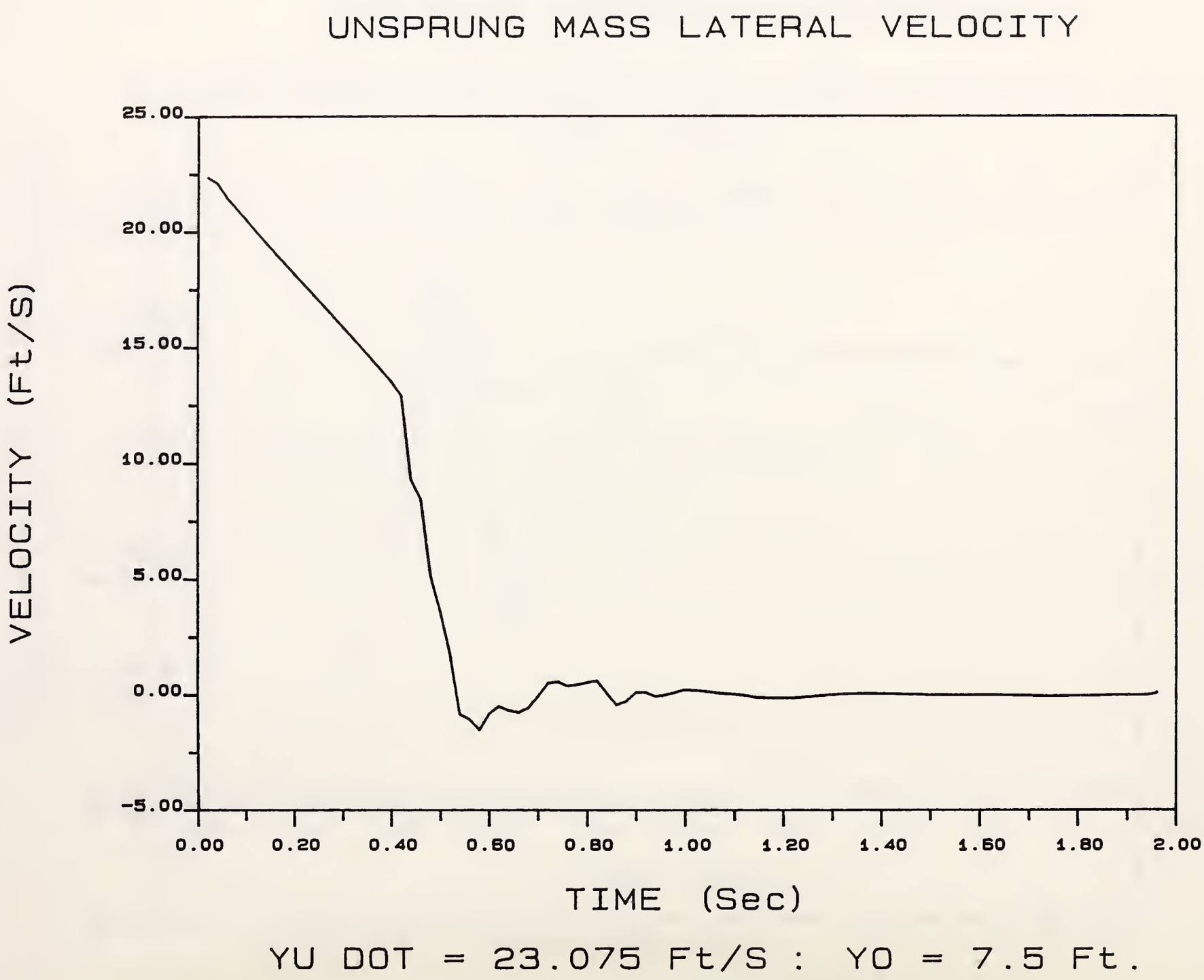
UNSPRUNG MASS LATERAL VELOCITY
VELOCITY (Ft/S)
25.00
20.00
15.00
10.00
5.00
0.00
-5.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YO = 7.5 Ft.

Fig. C-8

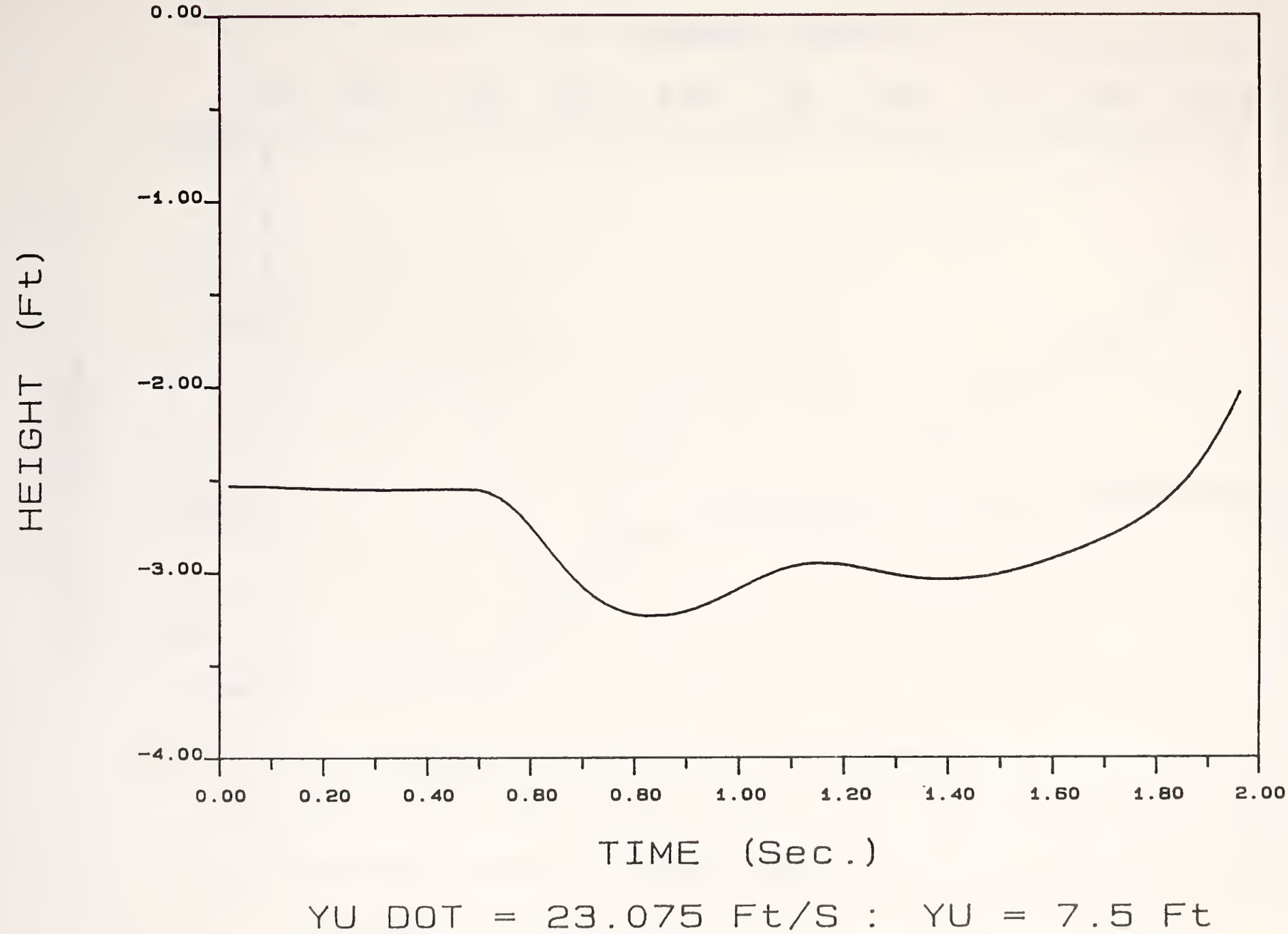
SPRUNG MASS C.G. POSITION
0.00
-1.00
-2.00
-3.00
-4.00
HEIGHT (Ft)
0.00
0.20
0.40
0.80
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec.)
YU DOT = 23.075 Ft/S : YU = 7.5 Ft

Fig. C-9

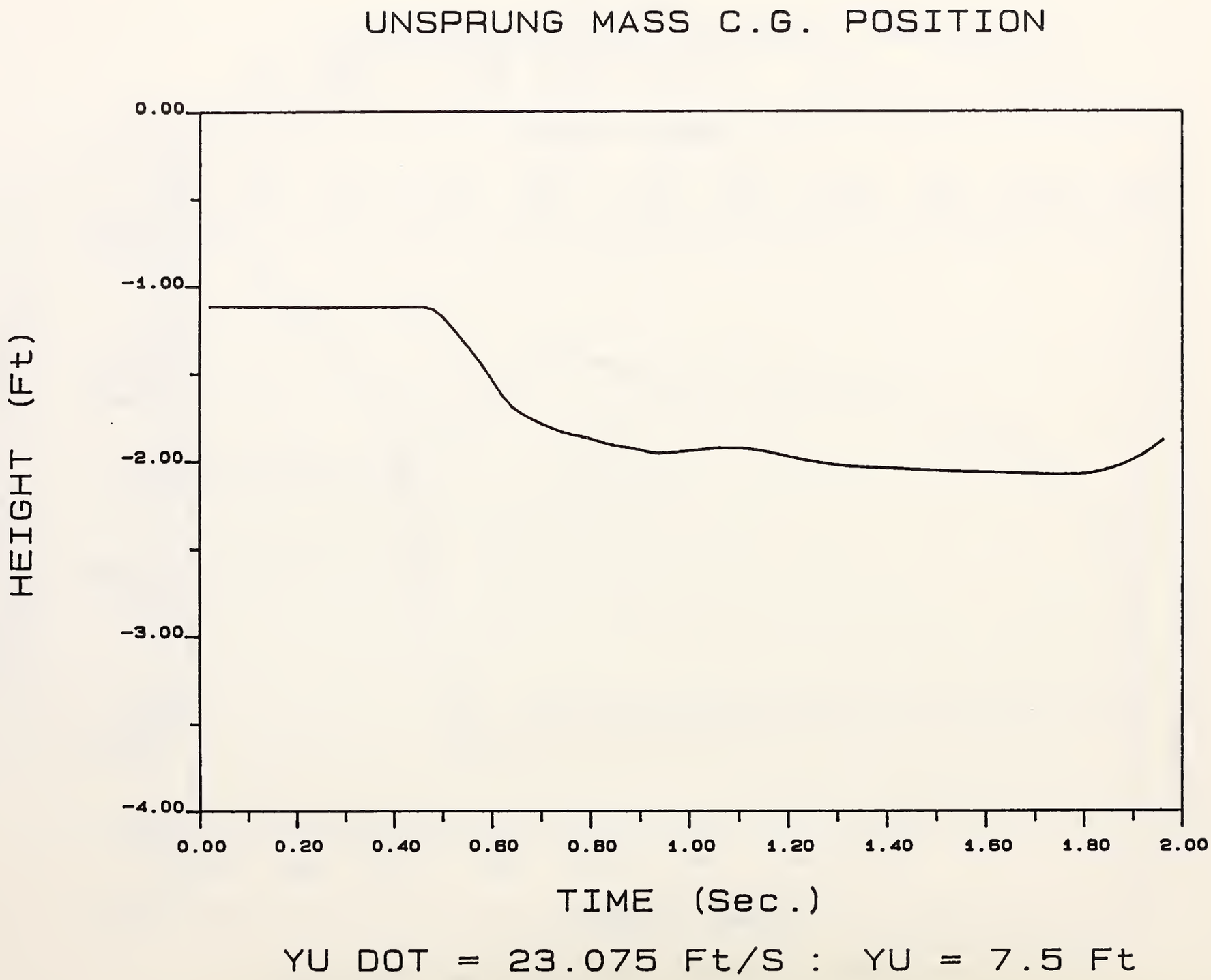
UNSPRUNG MASS C.G. POSITION
0.00
-1.00
-2.00
-3.00
-4.00
HEIGHT (Ft)
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec.)
YU DOT = 23.075 Ft/S : YU = 7.5 Ft

Fig. C-10

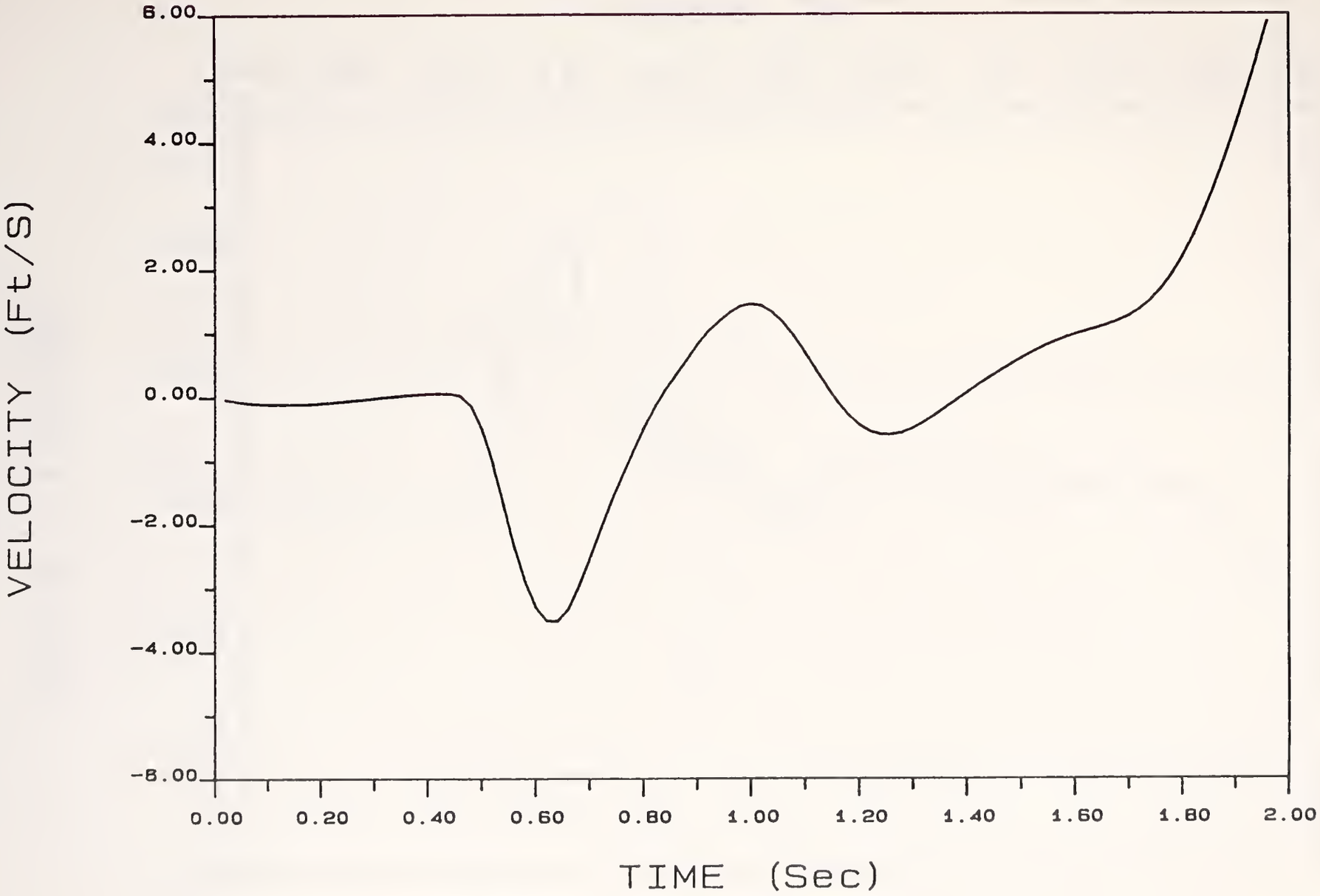
SPRUNG MASS VERTICAL VELOCITY
VELOCITY (Ft/S)
6.00
4.00
2.00
0.00
-2.00
-4.00
-6.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YO = 7.5 Ft.

Fig. C-11

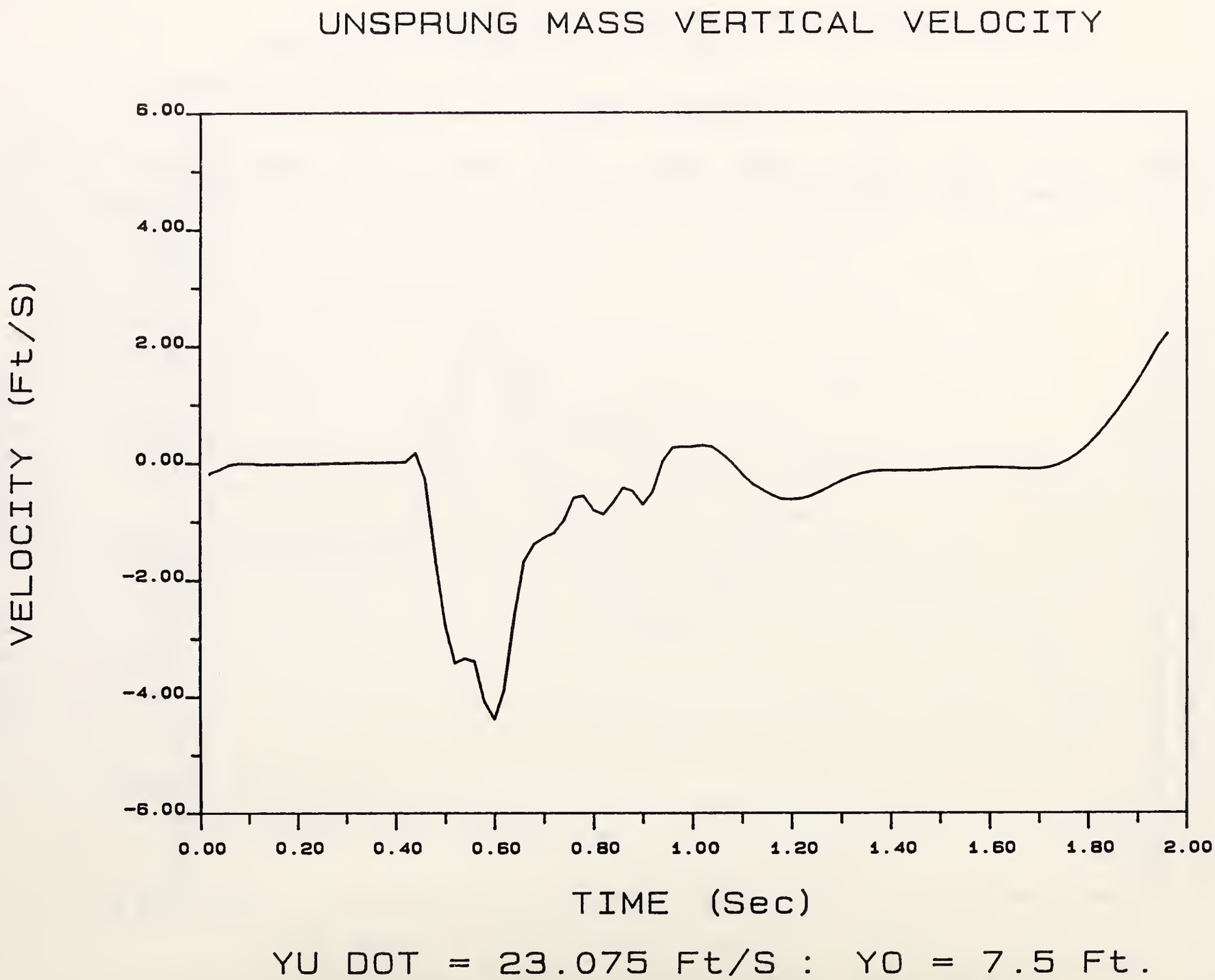
UNSPRUNG MASS VERTICAL VELOCITY
VELOCITY (Ft/S)
6.00
4.00
2.00
0.00
-2.00
-4.00
-6.00
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YO = 7.5 Ft.

Fig. C-12

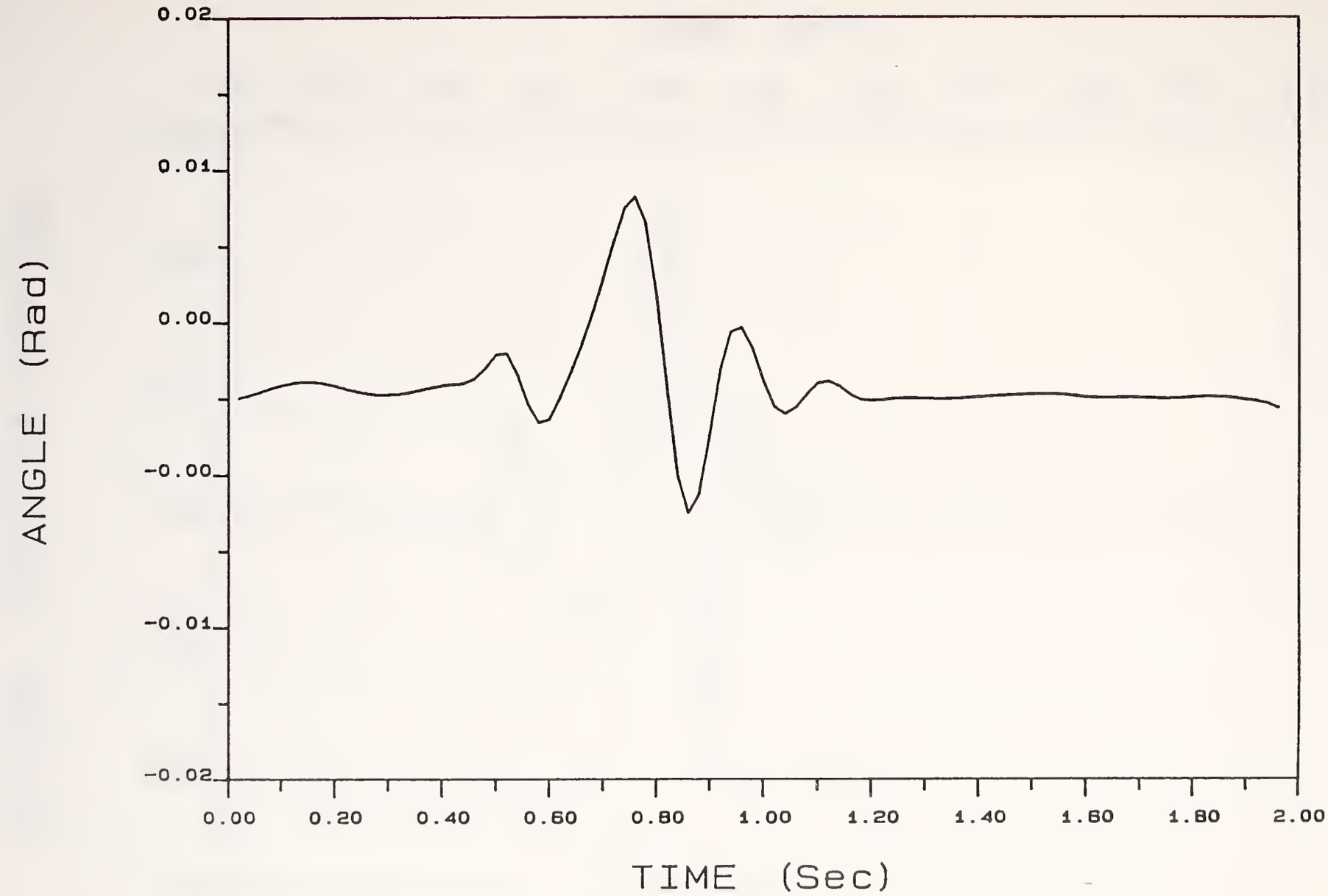
VEHICLE YAW ANGLE
ANGLE (Rad)
0.02
0.01
0.00
-0.00
-0.01
-0.02
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YO = 7.5 Ft.

Fig. C-13

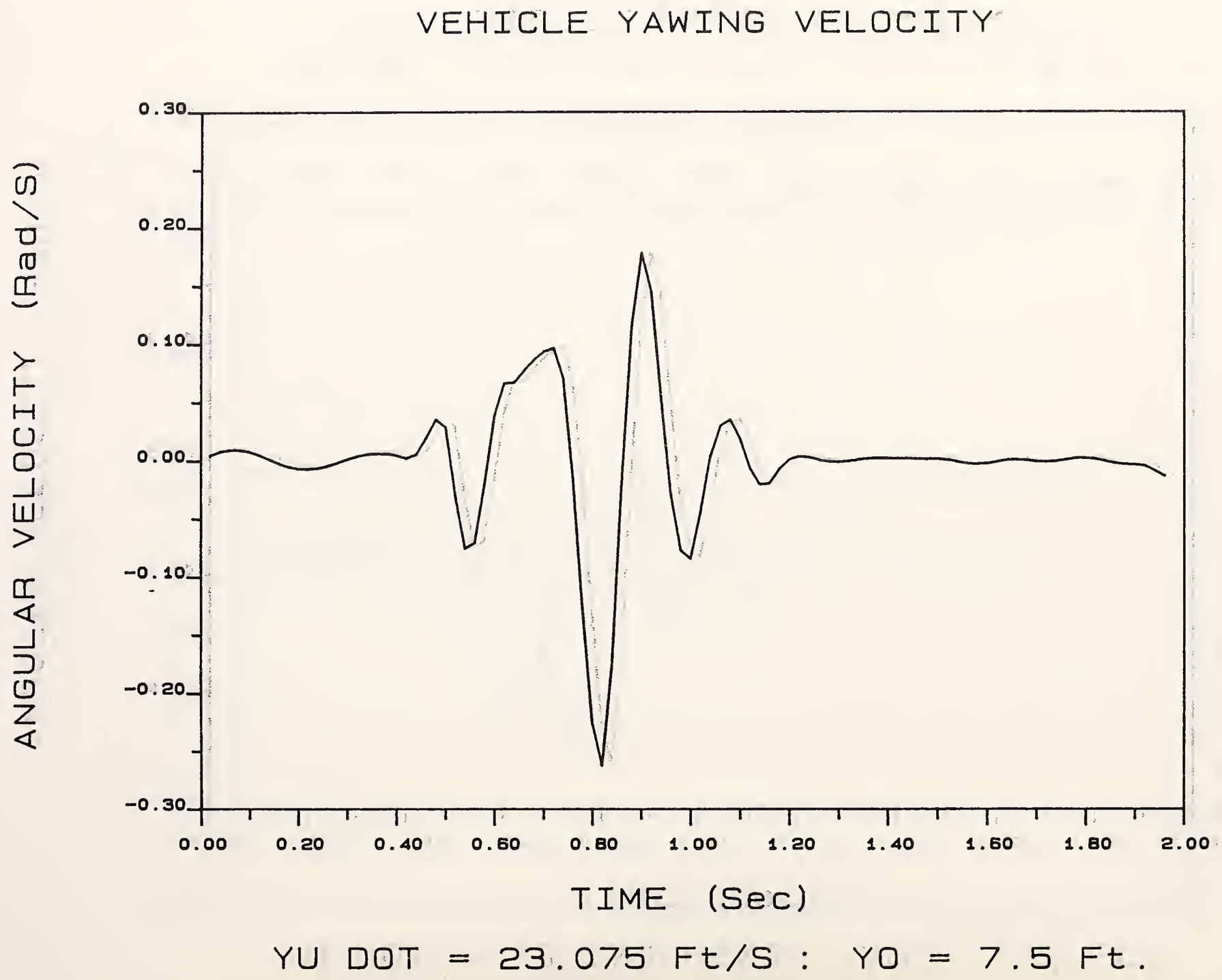
VEHICLE YAWING VELOCITY
ANGULAR VELOCITY (Rad/S)
0.30
0.20
0.10
0.00
-0.10
-0.20
-0.30
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YO = 7.5 Ft.

Fig. C-14

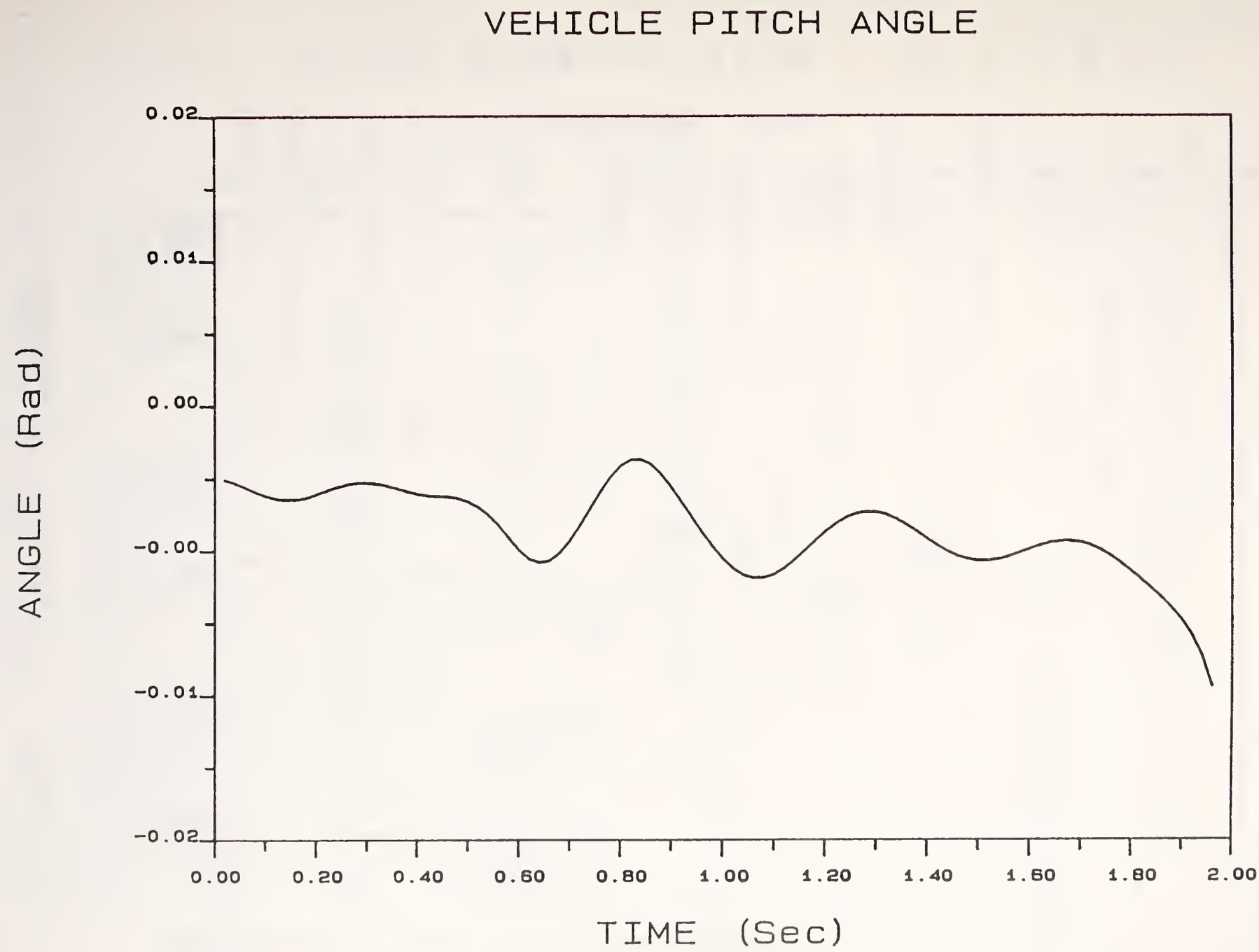
VEHICLE PITCH ANGLE
ANGLE (Rad)
0.02
0.01
0.00
-0.00
-0.01
-0.02
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YO = 7.5 Ft.

Fig. C-15

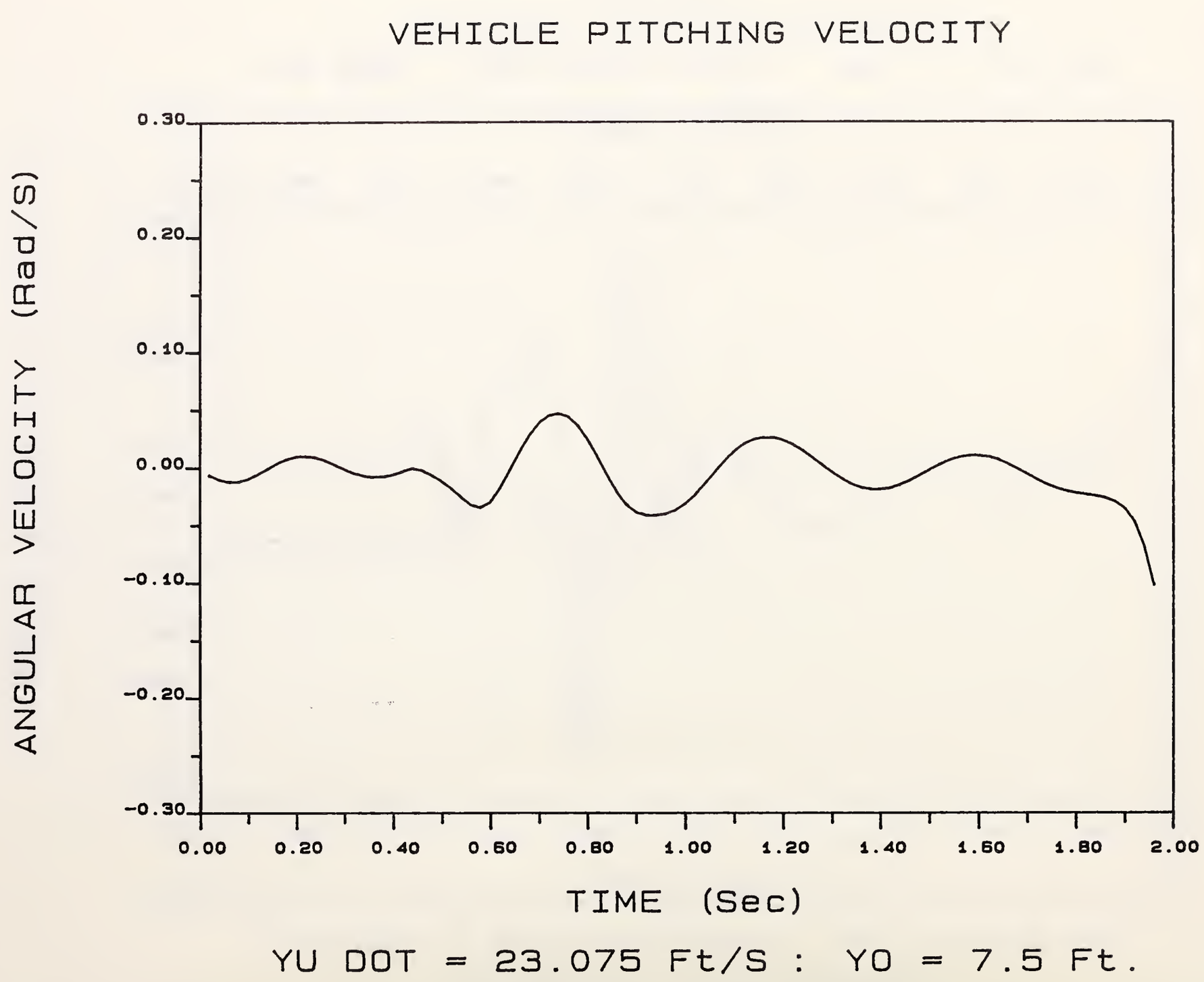
VEHICLE PITCHING VELOCITY
ANGULAR VELOCITY (Rad/S)
0.30
0.20
0.10
0.00
-0.10
-0.20
-0.30
0.00
0.20
0.40
0.60
0.80
1.00
1.20
1.40
1.60
1.80
2.00
TIME (Sec)
YU DOT = 23.075 Ft/S : YO = 7.5 Ft.

Fig. C-16

APPENDIX D: Fortran Source Code of STI's Tripped Vehicle Rollover Model

```
C ************************************************************
C *                                                          *
C *     SUBROUTINE NAME: MODEL                               *
C *                                                          *
C *     PURPOSE: 8 DOF NON-LINEAR MODEL BASED ON SYSTEMS     *
C *              TECHNOLOGY INC. ROLL OVER MODEL             *
C *                                                          *
C *     CALLED BY SUBROUTINE: FCN                            *
C *                                                          *
C *     LAST REVISION -  4/28/88        BY - ANDRZEJ G. NALECZ *
C *                                          ALAN BINDEMANN  *
C *                                                          *
C *     CALLING SEQUENCE:                                    *
C *                                                          *
C *             CALL MODEL(N,X,Y,YPRIME,VAR)                 *
C *                                                          *
C *     ARGUMENTS:                                           *
C *                                                          *
C *         INPUT: COMMON BLOCKS VDAT, AND SECP              *
C *                                                          *
C *        OUTPUT: INTEGRATION ARGUMENTS D(1-16)             *
C *                                                          *
C ************************************************************
C
      SUBROUTINE MODEL(N,X,Y,YPRIME,VAR,FLG1)
      IMPLICIT DOUBLE PRECISION (A-H,O-Z)
C
      COMMON/VDAT/    A,      L,      TRW,    HU,     KZ,     BZ,
     1  CF,     BD,     KD1,    KD2,    MU,     HR,     SUSK1,  SUSK2,
     2  B1,     B2,     IX,     IY,     IZ,     HS,     IU,     WHLDEF,
     3  KV,     POINT1, POINT2, MS,     IS,     TRAD,   HCURB,  S,
     4  SPRNGK, SPRLNG, ABSDEF, BL,     Y0,     DUM,    PC1,    PC2,
     5  PC3,    PC4,    PC5,    PC6,    PC7,    PC8,    PC9,    PC10,
     6  PC11
C
      COMMON/STATUS/ IP, INM
C
      COMMON/SECP/ SRSF, SPCF, DAMF, SPCR, DAMR,  ZSP,  ZUP, ZSDP,
     1             ZUDP, YSDP, YUDP
C
      INTEGER*4 N,IP,INM
C
      REAL*8  MS,     MU,     IS,     IU,     IX,     IY,     IZ,
     1        KZ,     KV,     ISA,    L,      KD1,    KD2,    IUA,
     2        MAXTRW,         H4(50,7),       TWF(50,7),
     3        TWR(50,7),      CRIT(50,7),     TRWF(50,7),
     4        TRWR(50,7),     YINT2(50,7),    YFMAX(50,7),
     5        YRMAX(50,7),    MAXTRWF(50,7),  MAXTRWR(50,7),
     6        VAR(50),        YPRIME(17),     Y(17)
C
C (1) ASSIGN INITIAL VALUES
C (2) UPDATED PARA SENT BACK TO MODEL
```

```
C
      CALL VDATA
C
C ASSIGN VARIABLE NAMES
C
      YSD   = Y(1)
      ZSD   = Y(2)
      PHISD = Y(3)
      YUD   = Y(4)
      ZUD   = Y(5)
      PHIUD = Y(6)
      THED  = Y(7)
      YAWR  = Y(8)
      YS    = Y(9)
      ZS    = Y(10)
      PHIS  = Y(11)
      YU    = Y(12)
      ZU    = Y(13)
      PHIU  = Y(14)
      THE   = Y(15)
      PSI   = Y(16)
      YGL   = Y(17)
C
C CALCULATE BUMP STOP SIZES
C
      G = 32.17D0
      BSDEF=SPRLNG-MS*G/SUSK1/2.0D0
      BSSIZL = BSDEF - ABSDEF
      BSSIZU = BSDEF + ABSDEF
C
C CALCULATE INITIAL VARIABLES (FIRST MODEL EVALUATION ONLY)
C
      IF (X.EQ.0.0D0) THEN
         CRIT(IP,INM)    = DATAN(TRW/2.0D0/HU)
         TWF(IP,INM)     = TRW/2.0D0
         TWR(IP,INM)     = TRW/2.0D0
         TRWF(IP,INM)    = TRW/2.0D0
         TRWR(IP,INM)    = TRW/2.0D0
         MAXTRWF(IP,INM) = TRW/2.0D0
         MAXTRWR(IP,INM) = TRW/2.0D0
         PHIU = (MU*G*CF*(TRAD + HU)+MS*G*CF*(HS+TRAD))/KZ/TRW/TRW
         PHIS = (MS*G*CF*(HS - HR)/2.0D0/SUSK1/S/S) + PHIU
         ZUP  = ZU  * PC7
         ZS = ZUP - (HS - HU + BSDEF) * DCOS(PHIS)
         ZSP  = ZS  * PC6
         YS = YU + (ZUP - ZSP) * DTAN(PHIS)
         YSDP = YSD * PC10
         YUD = YSDP
         YUDP = YUD * PC11
         ZSDP = ZSD * PC8
         ZUDP = ZUD * PC9
         H4(IP,INM) = -(MS * ZSP + MU * ZUP) / (MS + MU)
         YINT2(IP,INM) = WHLDEF - KD2 * POINT2
      END IF
C
```

```
        ZSP  = ZS  * PC6
        ZUP  = ZU  * PC7
        ZSDP = ZSD * PC8
        ZUDP = ZUD * PC9
        YSDP = YSD * PC10
        YUDP = YUD * PC11
C
C CHECK FOR ROLLOVER OR POTENTIAL OVERFLOW IN DGEAR
C
        IF (INM.Eq.1) THEN
          IF (DABS(PHIU).GT.CRIT(IP,INM)) THEN
            WRITE(5,105) PHIU * 57.3D0
105         FORMAT(1X,'PHIU = ',F7.3,' DEGREES')
            WRITE(5,*) 'UNSPRUNG MASS ROLL ANGLE > THAN CRIT. ANGLE'
            WRITE(5,*) 'ROLLOVER ASSUMED'
            STOP
          END IF
        ELSE IF (DABS(PHIU).GT.1.5708D0) THEN
           WRITE(5,106) PHIU * 57.3D0
106        FORMAT(1X,'PHIU = ',F7.3,' DEGREES DURING PARAM. VARIATION')
           WRITE(5,107) IP,INM
107        FORMAT(1X,'IP = ',I2,'  INM = ',I2)
           WRITE(5,*) 'COMPUTATIONS STOPPED TO PREVENT OVERFLOW IN DGEAR'
           STOP
        END IF
C
        SINPSI  = -DSIN(PSI)
        COSPSI  =  DCOS(PSI)
        SINPHIU =  DSIN(PHIU)
        COSPHIU =  DCOS(PHIU)
        TANTHE  =  DTAN(THE)
        COSTHE  =  DCOS(THE)
        COS2    =  COSPHIU * COSPHIU
        COS2THE =  COSTHE * COSTHE
C
        B=L-A
C
C CALCULATE CRUSH DISTANCE IN Y DIRECTION
C
        YF=(Y0-YGL)/COSPSI                ! UMC DEFORMATION LOGIC
        YR=YF+(A+B)*DTAN(PSI)             ! UMC DEFORMATION LOGIC
C
C CALCULATE RATE OF DEFORMATION
C
        YFD=(Y0-YGL)*YAWR*DTAN(PSI)/COSPSI-YUDP-YAWR*(A-        ! UMC LOGIC
     1       TRW/2.0D0*DTAN(PSI))
C
        YRD=YFD+(A+B)*YAWR/COSPSI/COSPSI                        ! UMC LOGIC
C
        IF (COSPSI.LT.0.0D0) THEN         ! UMC DEFORMATION LOGIC
           YF=-YF                         ! UMC DEFORMATION LOGIC
           YR=-YR                         ! UMC DEFORMATION LOGIC
           YFD=-YFD                       ! UMC DEFORMATION LOGIC
           YRD=-YRD                       ! UMC DEFORMATION LOGIC
        END IF                            ! UMC DEFORMATION LOGIC
```

```
C
C LEFT FRONT TIRE
C
        ZT  = (ZUP - TRW/2.0D0 * SINPHIU)/COSPHIU + HU - A * TANTHE
        ZTD = ZUDP/COSPHIU - PHIUD * (TRW/2.0D0 - ZUP*SINPHIU)/COS2
     1        - A * THED/COS2THE
        ZTLF = ZT
        ZTDLF=ZTD
        FZT=KZ*(TRAD+ZT)+BZ*ZTD
        IF (FZT.LE.0.0D0) GOTO 467
        GOTO 468
467     FZT = 0.0D0
468     CONTINUE
        FZLF = FZT
        DIST = HU + DABS(ZT)
        COSBETA=DCOS(1.5707-DABS(PHIU)-DATAN(DIST/(TRW/2.0D0)))
        ROLL = DSQRT(DIST*DIST + TRW*TRW*0.25D0) * COSBETA
        VEL=YUDP+A*YAWR-PHIUD*ROLL
        FYV=VEL/KV
        IF (VEL.GT. KV) FYV =  1.0D0
        IF (VEL.LT.-KV) FYV = -1.0D0
        FYT = CF * FZT * FYV
        FYLF = FYT
C
C RIGHT FRONT TIRE
C
        ZT  = ( ZUP + TRWF(IP,INM)*SINPHIU )/COSPHIU + HU - A*TANTHE
        ZTD = ZUDP/COSPHIU + PHIUD * (TRWF(IP,INM) + ZUP * SINPHIU)/
     1        COS2 - A * THED/COS2THE
        ZTRF = ZT
        ZTDRF = ZTD
        FZT=KZ*(TRAD+ZT)+BZ*ZTD
        IF (FZT.LE.0.0D0) GOTO 1467
        GOTO 1468
1467    FZT = 0.0D0
1468    CONTINUE
        FZRF = FZT
        DIST = HU + DABS(ZT)
        COSBETA = DCOS(1.5707 - DABS(PHIU) - DATAN(DIST/TRWF(IP,INM)))
        ROLL = DSQRT(DIST*DIST + TRWF(IP,INM)*TRWF(IP,INM)) * COSBETA
        VEL = YUDP + A*YAWR - PHIUD*ROLL
        FYV=VEL/KV
        IF (VEL.GT. KV) FYV =  1.0D0
        IF (VEL.LT.-KV) FYV = -1.0D0
        FYT = CF * FZT * FYV
        FYRF = FYT
C
C CALCULATE THE FRONT RIGHT DEFORMATION FORCE
C
C       YF=Y0-YU+A*SINPSI-TRW/2.0D0*COSPSI              ! STI LOGIC
C       YFD=-YUDP+A*YAWR*COSPSI+TRW/2.0D0*YAWR*SINPSI   ! STI LOGIC
C
        DUMY=YF
        DYMAX=YFMAX(IP,INM)
        MAXTRW=MAXTRWF(IP,INM)
```

```
        DTRW=TRWF(IP,INM)
C
        ABD = 1.0
        IF (DUMY.GE.0.0D0) GOTO 501
        DTRW = DUMY+TRW/2.0D0                          ! UMC DEFORMATION LOGIC
        DDUMY = -DUMY                                  ! UMC DEFORMATION LOGIC
C       DTRW = ((Y0-YU+A*SINPSI)/COSPSI)               ! STI DEFORMATION LOGIC
C       DDUMY = TRW/2.0D0-DTRW                         ! STI DEFORMATION LOGIC
        IF (DDUMY.LT.DYMAX) GOTO 510
        DYMAX = DDUMY
        IF (DDUMY.GT.POINT1) GOTO 496
        DUMYF = KD1 * DABS(DDUMY)
        GOTO 518
496     IF (DDUMY.GT.POINT2) GOTO 499
        DUMYF = WHLDEF
        GOTO 518
499     DUMYF = KD2 * DABS(DDUMY) + YINT2(IP,INM)
        GOTO 518
501     DUMYF = 0.0D0
        ABD = 0.0D0
        GOTO 521
C
C FRONT CURB FORCE AFTER MAXIMUM DEFORMATION
C
510     IF (DYMAX.LT.POINT2) GOTO 513
        DUMYF = KD2 * DABS(DDUMY) + YINT2(IP,INM)
        GOTO 517
513     IF (DYMAX.LT.POINT1) GOTO 516
        DUMYF = WHLDEF - KD1 * DABS(DYMAX-DDUMY)
        GOTO 517
516     DUMYF = KD1 * DABS(DDUMY)
517     IF (DUMYF.LT.0.0D0) THEN
           DUMYF = 0.0D0
           ABD = 0.0D0
        END IF
518     IF (DTRW.GT.MAXTRW) GOTO 520
        MAXTRW = DTRW
520     DTRW = MAXTRW
521     TRWDUM = MAXTRW
C
        YFMAX(IP,INM) = DYMAX
        TRWF(IP,INM) = DTRW
        TWF(IP,INM) = TRWDUM
        MAXTRWF(IP,INM) = MAXTRW
        SPCF = DUMYF * PC2
        DAMF = BD * ABD * YFD * PC3
        FDF = SPCF - DAMF
C
C LEFT REAR TIRE
C
        ZT  = (ZUP-TRW/2.0D0*SINPHIU)/COSPHIU + HU + B*TANTHE
        ZTD = ZUDP/COSPHIU - PHIUD*(TRW/2.0D0 - ZUP*SINPHIU)/COS2 +
     1        B * THED/COS2THE
        ZTLR = ZT
        ZTDLR = ZTD
```

```
        FZT = KZ * (TRAD + ZT) + BZ * ZTD
        IF (FZT.LE.0.0D0) GOTO 2467
        GOTO 2468
2467    FZT = 0.0D0
2468    CONTINUE
        FZLR = FZT
        DIST = HU + DABS(ZT)
        COSBETA = DCOS(1.5707 - DABS(PHIU) - DATAN(DIST/(TRW/2.0D0)))
        ROLL = DSQRT(DIST*DIST + TRW*TRW*0.25D0) * COSBETA
        VEL = YUDP - B*YAWR - PHIUD*ROLL
        FYV=VEL/KV
        IF (VEL.GT. KV) FYV =  1.0D0
        IF (VEL.LT.-KV) FYV = -1.0D0
        FYT = CF * FZT * FYV
        FYLR = FYT
C
C RIGHT REAR TIRE
C
        ZT  = ( ZUP + TRWR(IP,INM)*SINPHIU )/COSPHIU + HU + B*TANTHE
        ZTD = ZUDP/COSPHIU + PHIUD * (TRWR(IP,INM) + ZUP * SINPHIU)/
     1        COS2 + B * THED/COS2THE
        ZTRR = ZT
        ZTDRR = ZTD
        FZT=KZ*(TRAD+ZT)+BZ*ZTD
        IF (FZT.LE.0.0D0) GOTO 3467
        GOTO 3468
3467    FZT = 0.0D0
3468    CONTINUE
        FZRR = FZT
        DIST = HU + DABS(ZT)
        COSBETA = DCOS(1.5707 - DABS(PHIU) - DATAN(DIST/TRWR(IP,INM)))
        ROLL = DSQRT(DIST*DIST + TRWR(IP,INM)*TRWR(IP,INM)) * COSBETA
        VEL = YUDP - B*YAWR - PHIUD*ROLL
        FYV=VEL/KV
        IF (VEL.GT. KV) FYV =  1.0D0
        IF (VEL.LT.-KV) FYV = -1.0D0
        FYT = CF * FZT * FYV
        FYRR = FYT
C
C CALCULATE THE RIGHT REAR DEFORMATION FORCE
C
C       YR=Y0-YU-B*SINPSI-TRW/2.0D0*COSPSI                ! STI LOGIC
C       YRD=-YUDP-B*YAWR*COSPSI+TRW/2.0D0*YAWR*SINPSI     ! STI LOGIC
C
        DUMY=YR
        DYMAX=YRMAX(IP,INM)
        MAXTRW=MAXTRWR(IP,INM)
        DTRW=TRWR(IP,INM)
C
        ABD = 1.0
        IF (DUMY.GE.0.0D0) GOTO 701
        DTRW = DUMY+TRW/2.0D0                    ! UMC DEFORMATION LOGIC
        DDUMY = -DUMY                            ! UMC DEFORMATION LOGIC
C       DTRW = ((Y0-YU-B*SINPSI)/COSPSI)         ! STI DEFORMATION LOGIC
C       DDUMY = TRW/2.0D0-DTRW                   ! STI DEFORMATION LOGIC
```

```
        IF (DDUMY.LT.DYMAX) GOTO 710
        DYMAX = DDUMY
        IF (DDUMY.GT.POINT1) GOTO 696
        DUMYF = KD1 * DABS(DDUMY)
        GOTO 718
696     IF (DDUMY.GT.POINT2) GOTO 699
        DUMYF = WHLDEF
        GOTO 718
699     DUMYF = KD2 * DABS(DDUMY) + YINT2(IP,INM)
        GOTO 718
701     DUMYF = 0.0D0
        ABD = 0.0D0
        GOTO 721
C
C REAR CURB FORCE AFTER MAXIMUM DEFORMATION
C
710     IF (DYMAX.LT.POINT2) GOTO 713
        DUMYF = KD2 * DABS(DDUMY) + YINT2(IP,INM)
        GOTO 717
713     IF (DYMAX.LT.POINT1) GOTO 716
        DUMYF = WHLDEF - KD1*DABS(DYMAX-DDUMY)
        GOTO 717
716     DUMYF = KD1 * DABS(DDUMY)
717     IF (DUMYF.LT.0.0D0) THEN
           DUMYF = 0.0D0
           ABD = 0.0D0
        END IF
718     IF (DTRW.GT.MAXTRW) GOTO 720
        MAXTRW = DTRW
720     DTRW = MAXTRW
721     TRWDUM = MAXTRW
C
        YRMAX(IP,INM) = DYMAX
        TRWR(IP,INM) = DTRW
        TWR(IP,INM) = TRWDUM
        MAXTRWR(IP,INM) = MAXTRW
        SPCR = DUMYF * PC4
        DAMR = BD * ABD * YRD * PC5
        FDR = SPCR - DAMR
C
C CALCULATE SPRING FORCES
C
        DPHIS   = PHIS - PHIU
        DPHISD  = PHISD - PHIUD
        COSPHIS = DCOS(PHIS)
        SINPHIS = DSIN(PHIS)
        CDPHIS  = DCOS(DPHIS)
        SDPHIS  = DSIN(DPHIS)
        TDPHIS  = DTAN(DPHIS)
        CDPHIS2 = CDPHIS * CDPHIS
C
        DELZ  = ZUP - ZSP
        DELZD = ZUDP  - ZSDP
        DELY  = YS  - YU
        DELYD = YSDP  - YUDP
```

```
C
        DELSQ  = DSQRT(DELY*DELY + DELZ*DELZ)
        DELSQD = (DELZ*DELZD + DELY*DELYD) / DELSQ
C
C CALCULATE LEFT SPRING AND BUMP STOP FORCE
C
        SLL   = (DELSQ+HU) / CDPHIS - HS + S * TDPHIS
        SLLD  = DELSQD/CDPHIS + DPHISD*((DELSQ+HU)*SDPHIS+S)/CDPHIS2
        DEFLL = SPRLNG-SLL
        IF (SLL.LE.BSSIZL) GOTO 553
        IF (SLL.GE.BSSIZU) GOTO 555
        AK2 = 0.0D0
        AB2 = 0.0D0
        GOTO 558
553     AK2 = 1.0D0
        AB2 = 1.0D0
        DBL = BSSIZL - SLL
        GOTO 556
555     DBL = BSSIZU - SLL
556     DBLD = SLLD
C
558     FSL = SUSK1 * DEFLL - B1 * SLLD +
     1        AK2 * SUSK2 * DBL - AB2 * B2 * DBLD
C
C CALCULATE RIGHT SPRING AND BUMP STOP FORCE
C
        SLR  = (DELSQ+HU) / CDPHIS - HS - S * TDPHIS
        SLRD = DELSQD/CDPHIS + DPHISD*((DELSQ+HU)*SDPHIS-S)/CDPHIS2
        DEFLR = SPRLNG - SLR
        IF (SLR.LE.BSSIZL) GOTO 572
        IF (SLR.GE.BSSIZU) GOTO 574
        AK2 = 0.0D0
        AB2 = 0.0D0
        GOTO 577
572     AK2 = 1.0D0
        AB2 = 1.0D0
        DBR = BSSIZL - SLR
        GOTO 575
574     DBR = BSSIZU - SLR
575     DBRD = SLRD
C
577     FSR = SUSK1 * DEFLR - B1 * SLRD +
     1        AK2 * SUSK2 * DBR - AB2 * B2 * DBRD
C
        SDPHIU = DSIN(PHIU-PHIS)
        CDPHIU = DCOS(PHIU-PHIS)
        DPHIUD = PHIUD-PHISD
        DELP   = DELY*COSPHIS - DELZ*SINPHIS - (HR-HU)*SDPHIU
        DELPD1 = PHISD * (DELZ*COSPHIS + DELY*SINPHIS)
        DELPD2 = DELYD * COSPHIS - DELZD * SINPHIS
        DELPD3 = (HR - HU) * CDPHIU * DPHIUD
        DELPD  = DELPD2 - DELPD1 - DELPD3
C
        PINF1 = DELP*SPRNGK + BL*DELPD
C
```

```
C CALCULATE THE INTEGRATION ARGUMENTS
C
      YSDD = ((FSL+FSR)*SINPHIS - PINF1*COSPHIS)/MS
C
      ZSDD = (-(FSL+FSR)*COSPHIS - PINF1*SINPHIS)/MS + G
C
      HP1 = DELY - HR*SINPHIU
      HP2 = DELZ - HR*COSPHIU
      HPINF1 = DSQRT(HP1*HP1 + HP2*HP2)
C
      PHISDD = ((FSL-FSR) * S + PINF1 * HPINF1) / IS
C
      FYL = FYLF + FYLR
      FYR = FYRF + FYRR
      FZL = FZLF + FZLR
      FZR = FZRF + FZRR
C
      YUDD = (PINF1*COSPHIS-(FSL+FSR)*SINPHIS-(FYR+FYL+FDF+FDR))/MU
C
      ZUDD = ((FSL+FSR)*COSPHIS+PINF1*SINPHIS-FZL-FZR)/MU + G
C
      FREDLF = (HU + DABS(ZTLF)) * SINPHIU
      FREDRF = (HU + DABS(ZTRF)) * SINPHIU
      FREDLR = (HU + DABS(ZTLR)) * SINPHIU
      FREDRR = (HU + DABS(ZTRR)) * SINPHIU
      BOB1 = -ZUP - HCURB
C
      TERM1 = (FSR-FSL) * S * DCOS(DPHIS)
      TERM3 = FZLF * (TRW/2.0D0 * COSPHIU + FREDLF)+
     1        FZLR * (TRW/2.0D0 * COSPHIU + FREDLR)
      TERM4 = FZRF * (TWF(IP,INM) * COSPHIU - FREDRF)+
     1        FZRR * (TWR(IP,INM) * COSPHIU - FREDRR)
      TERM5 = FYL * ZUP
      TERM6 = FYR * ZUP
      TERM7 = (HR - HU) * PINF1 * DCOS(DPHIS)
      TERM9 = FDF * BOB1 + FDR * BOB1
C
      PHIUDD = (TERM1+TERM3-TERM4-TERM5-TERM6+TERM7+TERM9)/IU
C
      THEDD = ((IZ-IX)*YAWR*PHISD - B*(FZLR+FZRR)+A*(FZLF+FZRF))/IY
C
      YAWRD = ((FYLR + FYRR + FDR)*B - (FYLF + FYRF + FDF)*A +
     1          PHISD * THED * (IX - IY)) / IZ
C
      YPRIME(1) = YSDD
      YPRIME(2) = ZSDD
      YPRIME(3) = PHISDD
      YPRIME(4) = YUDD
      YPRIME(5) = ZUDD
      YPRIME(6) = PHIUDD
      YPRIME(7) = THEDD
      YPRIME(8) = YAWRD
C
      YPRIME(9)  = YSD
      YPRIME(10) = ZSD
```

```
      YPRIME(11) = PHISD
      YPRIME(12) = YUD
      YPRIME(13) = ZUD
      YPRIME(14) = PHIUD
      YPRIME(15) = THED
      YPRIME(16) = YAWR
      YPRIME(17) = YUD*COSPSI+YAWR*(A*COSPSI+
     1             TRW/2.0D0*SINPSI)
C
C     SET VARIABLE VALUES EQUAL TO VALUES CALCULATED BY SUBROUTINE
C     MODEL
C
      VAR(18) = V
      VAR(19) = FZL
      VAR(20) = YPRIME(3)
      VAR(21) = YPRIME(6)
C
C     CALCULATE THE MASS MOMENTS OF INERTIA AROUND PT A
C
      YUA = (TRAD + HU) * SINPHIU -
     1      (TRWF(IP,INM) + TRWR(IP,INM)) / 2.0D0 * COSPHIU
      YSA = YUA + (YS - YU)
      ISA = IS + MS * ( (ZSP + HCURB) * (ZSP + HCURB) + YSA*YSA )
      IUA = IU + MU * ( (ZUP + HCURB) * (ZUP + HCURB) + YUA*YUA )
C
C     CALCULATE THE INSTANTANEOUS AVG C.G. HEIGHT
C
      H3 = -(MS*ZSP + MU*ZUP) / (MS+MU)
C
C     CALCULATE THE NONCENTRIODAL K.E.
C
      TNR = (ISA * PHISD*PHISD + IUA * PHIUD*PHIUD) / 2.0D0
      VAR(22) = TNR
C
C     CALCULATE THE GENERAL PLANE MOTION ENERGY
C
      TGPM = ( MS*(YSDP*YSDP + ZSDP*ZSDP) + MU*(YUDP*YUDP + ZUDP*ZUDP)
     1       + IS*PHISD*PHISD + IU*PHIUD*PHIUD ) / 2.0D0
C
      VAR(23) = TGPM
C
C     CALCULATE THE TRANSLATIONAL ENERGY
C
      TRAN = TGPM - TNR
      VAR(24) = TRAN
C
C     CALCULATE THE POTENTIAL ENERGY OF THE VEHICLE
C
      VENG = (MS + MU) * G * (H3 - H4(IP,INM))
      VAR(25) = VENG
C
C     CALCULATE THE TOTAL ENERGY OF THE SYSTEM
C
      VAR(26) = VENG + TGPM
C
```

```
C       CALCULATE THE CRITICAL ENERGY OF ROLOVER
C
        SRSF = TRW / 2.0D0 / H4(IP,INM) * PC1
        PCRIT = G * (MS + MU) * H4(IP,INM) * ( DSQRT(1.0D0 + SRSF**2)
     1             - 1.0D0 )
C
        VCRIT = PCRIT
C
        VAR(27) =  PCRIT - TNR
C
C  UPDATED SEC. PARA SENT  TO DGEAR
C
        CALL VDATA
C
C REASSIGN VARIABLE VALUES
C
        Y(1)  = YSD
        Y(2)  = ZSD
        Y(3)  = PHISD
        Y(4)  = YUD
        Y(5)  = ZUD
        Y(6)  = PHIUD
        Y(7)  = THED
        Y(8)  = YAWR
        Y(9)  = YS
        Y(10) = ZS
        Y(11) = PHIS
        Y(12) = YU
        Y(13) = ZU
        Y(14) = PHIU
        Y(15) = THE
        Y(16) = PSI
        Y(17) = YGL
C
        RETURN
        END
```

Data File for Sensitivity Program
1975 Nova

```
1,'DIST FROM C.G. TO FRONT AXLE          ',4.45D0
1,'WHEELBASE                             ',9.25D0
1,'VEHICLE TRACK WIDTH                   ',5.0D0
0,'DIST FROM AXLE TO UNSPRUNG MASS CG    ',0.2D0
0,'SINGLE TIRE COMPRESSION RATE CONST.   ',12000.0D0
0,'SINGLE TIRE DAMPING CONST.            ',80.0D0
0,'COEFF. OF FRICTION TIRE/GROUND        ',0.75D0
0,'CURB FORCE DAMPING CONSTANT           ',0.25D0
0,'SLOPE OF WHEEL COMPLIANCE AT CURB     ',24000.0D0
0,'SLOPE OF CAR DEFORMATION AT CURB      ',96000.0D0
0,'TOTAL UNSPRUNG MASS                   ',16.7D0
0,'DIST FROM AXLE TO LAT. REST. FORCE    ',1.0D0
1,'SUSPENSION SPRING CONST               ',6900.0D0
0,'BUMP STOP SPRING CONST                ',33000.0D0
1,'SUSPENSION DAMPING CONST.             ',325.0D0
0,'BUMP STOP DAMPING CONST.              ',33.0D0
0,'TOTAL ROLL INERTIA                    ',330.0D0
0,'TOTAL PITCH INERTIA                   ',2240.0D0
1,'TOTAL YAW INERTIA                     ',2240.0D0
0,'DIST FROM SPR. MASS BOTT. TO SMCG     ',1.2D0
0,'TOTAL UNSPRUNG MASS ROLL INERTIA      ',132.0D0
0,'CONSTANT FORCE OF WHEEL DEF. @ CURB   ',4000.0D0
0,'VEL. OF CONSTANT SIDE VEL. FUNCTION   ',1.0D0
0,'CRUSH DIST WHERE SUSPENSION DEFORMS   ',0.1667D0
0,'CRUSH DIST WHERE BODY DEFORMS         ',0.6667D0
1,'SPRUNG MASS                           ',105.0D0
1,'SPRUNG MASS ROLL INERTIA              ',240.0D0
1,'TIRE RADIUS                           ',1.0D0
0,'CURB HEIGHT                           ',0.5D0
0,'HALF DIST OF SPRING BASE              ',1.75D0
0,'LATERAL BUSHING SPRING CONST.         ',200000.0D0
0,'UNLOADED SUSPENSION SPRING LENGTH     ',0.6667D0
0,'DIST SS AXLE TO BUMP STOPS            ',0.25D0
0,'LATERAL BUSHING DAMPING CONSTANT      ',1000.0D0
0,'LATERAL POSITION OF CURB              ',7.50D0
0,'DUMMY TEST PARAMETER                  ',1.0D0
0,'ENDPP                                 ',0.0D0
```

```
0,'PC1  - SRSF                                    ',1.0D0
0,'PC2  - SPCF                                    ',1.0D0
0,'PC3  - DAMF                                    ',1.0D0
0,'PC4  - SPCR                                    ',1.0D0
0,'PC5  - DAMR                                    ',1.0D0
0,'PC6  - ZSP                                     ',1.0D0
0,'PC7  - ZUP                                     ',1.0D0
0,'PC8  - ZSDP                                    ',1.0D0
0,'PC9  - ZUDP                                    ',1.0D0
0,'PC10 - YSDP                                    ',1.0D0
0,'PC11 - YUDP                                    ',1.0D0
0,'ENDSP                                          ',0.0D0
0,'1  YS Dot                                      ',0.0D0
0,'2  ZS Dot                                      ',0.0D0
0,'3  PHIS Dot                                    ',0.0D0
0,'4  YU Dot                                      ',0.0D0
0,'5  ZU Dot                                      ',0.0D0
0,'6  PHIU Dot                                    ',0.0D0
0,'7  THETA Dot                                   ',0.0D0
0,'8  r                                           ',0.0D0
0,'9  YS                                          ',0.0D0
0,'10 ZS                                          ',0.0D0
0,'11 PHIS                                        ',0.0D0
0,'12 YU                                          ',0.0D0
1,'13 ZU                                          ',0.0D0
0,'14 PHIU                                        ',0.0D0
0,'15 THETA                                       ',0.0D0
0,'16 PSI                                         ',0.0D0
0,'17 Y GLOBAL                                    ',0.0D0
0,'18 V                                           ',0.0D0
0,'19 FZL                                         ',0.0D0
0,'20 PHISDD                                      ',0.0D0
0,'21 PHIUDD                                      ',0.0D0
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0,'27 RPER                                        ',0.0D0
0,'ENDOV                                          ',0.0D0
```

APPENDIX E

Parameter Set Description

Geometric Parameter Set #1:

Label	Description
TRW	Vehicle track width
h_u	Height of unsprung mass c.g. position above axle centerline
h_r	Height of lateral restraining force member (roll axis) above the axle centerline (average front and rear)
h_s	Static height of sprung mass c.g. position above axle centerline
L	Vehicle wheelbase
a	Horizontal distance from the vehicle c.g. position to the front axle
SRSF	The ratio of track width to c.g. height

Geometric Parameter Set #2:

Label	Description
H_{curb}	Height of the curb
SPRLNG	The undeflected length of suspension springs
T_r	Undeformed tire radius
ABSDEF	Vertical wheel travel distance from the static position of the wheel to the point where contact with the suspension bump stop begins
s	Half of the distance between left and right suspension springs
Y_u	Initial distance between vehicle and curb (measured from front inside tire)

Mass Parameter Set:

Label	Description
M_s	Total sprung mass
M_u	Total unsprung mass
I_x	Total roll inertia about total mass center of gravity
I_Y	Total pitch inertia
I_Z	Total yaw inertia
I_s	Sprung mass roll inertia about its center of gravity
I_u	Total unsprung mass roll inertia about M_u center of gravity

Stiffness Parameter Set:

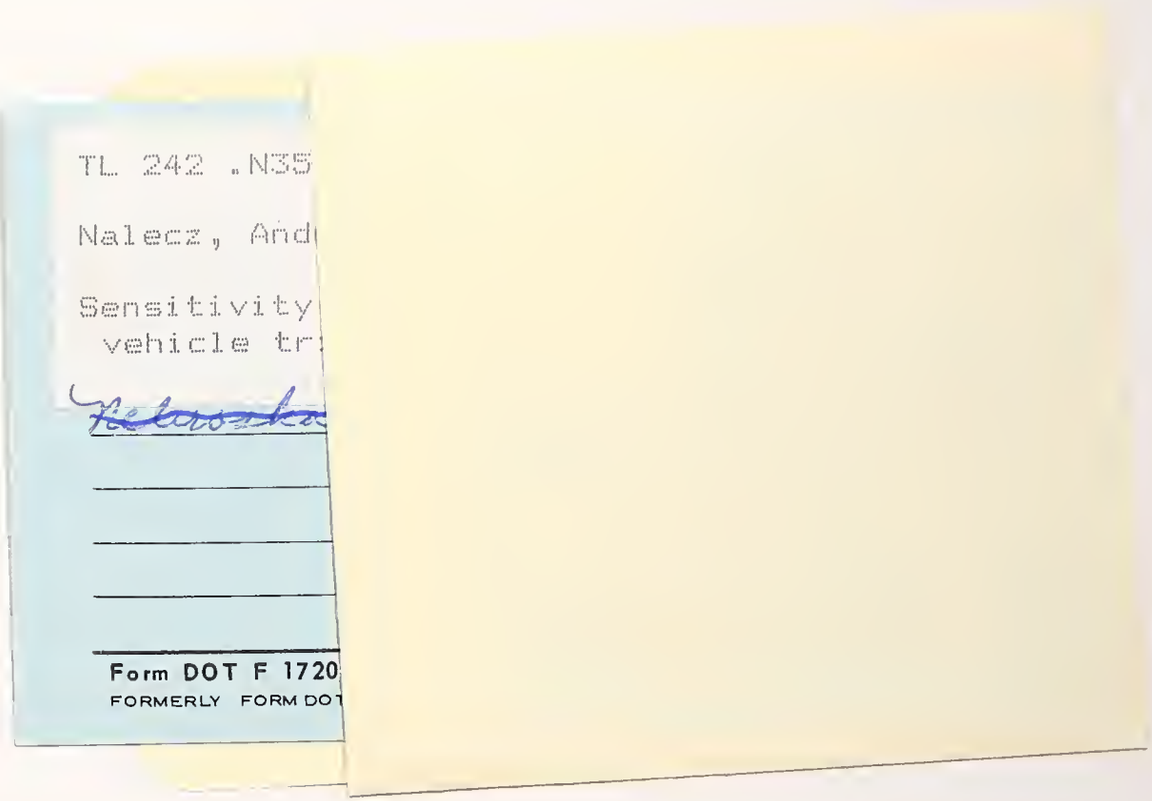
Label	Description
K_1	Suspension spring rate per side, equivalent acting on wheel
K_2	Bump stop springrate per side, equivalent acting at wheel
K	Bushing (lateral restraining force) spring constant
K_z	Single tire radial spring rate
KD1	Wheel and suspension lateral deflection spring rate when striking the curb
KD2	Slope of force vs. lateral deformation of body, frame and engine when striking the curb

Damping Parameter Set:

Label	Description
B_1	Suspension shock absorber damping rate per side, equivalent acting on wheel
B_2	Bump stop damping rate per side, equivalent acting at wheel
B	Bushing (lateral restraining force) damping constant
B_z	Single tire radial damping rate
B_d	Vehicle structure damping constant in the lateral deformation direction

Force Parameter Set:

Label	Description
μ	Coefficient of friction between tire/ground
WHLDEF	Constant force during crush deformation of the "soft" suspension and wheel members when striking a curb
K_v	Velocity at which tire slides with a constant side velocity function
POINT1	Deflection distance at which wheel and suspension begin to crush (permanent deformation)
POINT2	Crush distance at which the more solid members of the body, frame, and engine become involved in lateral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